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JOURNAL
OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XLI. NOVEMBER 1929. No. 4

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DIESEL ENGINES FOR MOTOR BOATS.

By LIEUTENANT JOHN O. HUSE, U. S. NAVY.*†

The frequent recurrence of motor boat fires with resulting loss of life and property is incontrovertible testimony to the fact that motors employing volatile fuels are inherently hazardous in motor boats.

These fires recur on gasoline engines, notwithstanding a long campaign waged by the Navy with intelligence, ingenuity, and energy. This effort was directed toward the safer design and operation of the gasoline engine. The campaign has obviously failed.

Assuming that the problem of motor boat fire prevention cannot be solved by the improvement and modification of the gaso-

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line engine, it is evident that, unless the frequent danger to life be accepted with equanimity, an effort must be made to develop a satisfactory engine employing less volatile fuels.

Other Naval Powers, impelled by the common motive, have turned to crude fuel engines. One is attempting to perfect a kerosene engine employing a carburetor, pre-combustion chamber, and heating device for cold starting.

The German Admiralty, however, encouraged by Diesel success in rail cars, busses, trucks, and construction machinery, have solved the problem by the adoption of high speed Diesel engines.

The following is, therefore, a plea for the fair trial of the gasoline engine versus the Diesel engine in the small boats of the Navy. The plea is based not only on the precedent set by the German Navy but also upon a thousand running engines; the confidence of their commercial operators; personal inspection of a dozen enthusiastic manufacturies; and the following description of the many high speed Diesel types which are on the market. The plea, consequently, is not fanciful or theoretical.

ADVANTAGES.

The compelling argument for the use of Diesel engines in our boats rests on the fact that its less combustible fuel and its elimination of the ignition circuit nearly *precludes the fire hazard.* An examination of the vaporization curves, vapor pressures, and flash points of Diesel oil and gasoline illustrate eloquently the lessened fire hazard on the Diesel motor boat.

The lower cost of fuel on the Diesel engine must not be overlooked, for this saving will pay for the expense of the Diesel installation. This saving as shown by several test results in Germany amounts to 70-76 per cent of gasoline fuel bill.

This saving is obtained by two factors:—

(*a*) The Diesel engine is a more economical engine *thermally* and the increased efficiency of the Diesel high compression engine over the Otto low compression engine is often as much as 25 per cent, and the cruising radius of the motor boat on the same fuel capacity is proportionately increased.

(*b*) In addition to the reduced weight of fuel demanded by the Diesel is added the *large price differential* between the cost of

Diesel fuel and the cost of gasoline. This difference is so great that in Germany, where gasoline costs are perhaps high, the combined saving frequently is as great as 75 per cent.

The elimination of the ignition circuits not only lessens the fire hazard but it also removes the troublesome breakdowns due to *wet magnetos*. This advantage is very important in motor boat engines which are only with great difficulty made moisture proof.

The qualities possessed by the Diesel engine of developing constant or even slightly increasing torque as the speed falls, with much better power pick-up, are both desirable but not as important, in the motor boat as in the motor vehicle.

DISADVANTAGES OF THE SMALL DIESEL IN MOTOR BOATS.

The first cost will always be much higher than that of the gasoline engine because closer tolerances, better workmanship, and better materials are necessary in the Diesel.

The Diesel motor, because of its higher compression and maximum pressures, is more subject to shock and mechanical vibration than the gasoline or lower compression engine.

The higher compression also causes the Diesel to be inherently harder to start than the gasoline motor.

The higher compression and maximum pressures involve heavier engine framing, connecting rods, and crankshafts. Fortunately for the small Diesel, casting and manufacturing limitations on gasoline engines, cause the wall thickness of the cylinders, cylinder heads, pistons, and framing to be made unnecessarily heavy and, in general, the same weights will support the higher pressures of the Diesel. The increase in Diesel weight need be only 10 to 15 per cent as compared to the gasoline engine.

Poor combustion at idling loads is sometimes encountered. This is unimportant on motor boats.

SIMILARITY BETWEEN MOTOR BOAT AND AUTOMOTIVE DIESEL ENGINES.

A motor boat Diesel is assumed to be a high speed Diesel engine approaching in power, weight, speed, and space requirement the gasoline engines in the motor boats in the Navy.

It is also assumed that any high speed auto truck, autobus, or railway car Diesel will be suitable for motor boats not only because the much more rigid demands in this service are similar to those in the motor boat, but also because the commercial types on sale in Europe are sold both for boat and automotive use.

Automotive Diesels are suitable for motor boat use.

Having discussed the undoubted advantages of the Diesel in the gasoline motor field, the natural query may be "Why, if the disadvantages are surmountable and do not outweigh the advantages, has the Diesel not entered the automotive and motor boat field?"

In 1894, when Dr. Rudolf Diesel developed the cycle, the internal combustion engine of greatest sales possibility was the slow gas engine for power stations. The mechanization of vehicles was in its infancy. The first designed Diesels were meant to supplant the gas engine. They were slow speed, large cylindered engines which by their sizes multiplied the mechanical troubles incident to high compression.

The tremendous advantage of the Diesel for high speed use was realized long before the World War. Small cylindered and faster running engines, were attempted. Design and financial failure followed these attempts. Gasoline engines had won the auto market and the sales resistance to Diesels was insurmountable. Success with fast running submarine engines was finally realized prior to, and during, the World War. These engines attained a speed of 500 R.P.M. and weight of but 40 pounds per horsepower. The common belief that Diesel engines were slow running, elephantine engines was finally dispelled and the development to the smaller engines of 1500 R. P. M. and but 15 pounds per horsepower for automotive use has rapidly progressed during the last ten years.

The technical factors which have contributed greatly to the progress of the automotive Diesel are:

(*a*) Increased use of the solid injection of fuel has won many friends for the auto Diesel because it has eliminated the complicated air compressor, continuous source of trouble and absorber of about 10 per cent of the power developed.

(*b*) Development of the aluminum piston to reduce reciprocating weights and inertia forces in turn permitting higher piston speeds, and hence lower weights per horsepower.

(*c*) The improvements in metals making possible the use of cast aluminum alloys, high grade nickel steel, thin section steel castings and high grade cast iron which will withstand the more rigid demands of the high compression motor and give it the necessary life and light weight.

The economic factors which have hastened its development in recent years are:

(*a*) The enormous expansion in the automotive sales field in Europe, which is more "Diesel minded" than America.

(*b*) The high price of gasoline and the excessive spread between the costs of Diesel fuel and gasoline fuel in Europe.

(*c*) The introduction of the economical Diesel in trucks, busses, and rail cars which, unlike pleasure cars, are dividend paying.

(*d*) The large number of competing European manufacturers of large Diesels selling in too restricted a sales field.

A discussion of high speed design discloses construction features common to all designs:

(*a*) Every design employs four cycle action, except Junkers.

(*b*) Every automotive design is single acting.

(*c*) All have framing with cylinders in line.

(*d*) All have overhead valves. (Junkers excepted.)

(*e*) Every design uses aluminum pistons for lightening reciprocating weights.

(*f*) All manufacturers use the solid injection of fuel except Maybach in Friedrichshafen.

The following discussion of the different types developed lays particular emphasis on those features in which the motor boat or automotive Diesel engine differs from the gasoline engine and on those features in design in which there exists a great difference between the designs of the different Diesel manufacturers:

INJECTION AND COMBUSTION.

Thermally and mechanically the automotive Diesel differs from the automotive gasoline engine largely in the manner of injection and combustion of the fuel. In order to inject and burn the

fuel in .002 to .003 seconds, extraordinary means must be adopted to inject the fuel, raise it to the ignition temperature, mix it with the air in the combustion chamber, and burn it completely.

The combustion problem is not as difficult in the smaller cylinder diameters as in the larger cylindered slower running engines. The volume of the combustion chamber is small. The length of the pipe lines is small. Less difficulty with injection delays is encountered.

The injection, atomization, distribution, and combustion is accomplished with the following types of fuel pumps, combustion chambers and injection nozzles.

FUEL PUMPS AND LOAD CONTROL.

The first consideration in the injection and combustion of the fuel should be the fuel pump and load control. In automotive engines, the two factors cannot be divorced because the load control is always obtained by the mechanism of the fuel pump. The success achieved by the small Diesel in recent years may be largely attributed to the great strides made recently in the design and manufacture of the fuel pump.

The functions of the fuel pump are as follows:

(*a*) Delivery of fuel oil to the injection nozzle at a high pressure.

(*b*) Measurement of this fuel oil to very fine limits.

The small Diesel fuel pump is a much more difficult problem than the pump for larger slower running Diesels. The reasons are quite evident when it is considered that some engines when idling require pumps which will deliver only 10 cubic millimeters, or .0006102 cubic inch by volume.

The fuel pumps must deliver this infinitesimal quantity of oil to the nozzle at a high pressure, timed to .0001 second, and at a constant rate. Each cylinder must receive the same quantity of oil or serious load unbalance results. The usual accuracy required is 4 to 5 per cent load unbalance at full load and many pumps run as close as 2 per cent.

It is not to be wondered that great ingenuity has been expended in the design and manufacture of such a delicate mechanism. Upon

the simplicity, ruggedness and efficiency of the fuel pumps, hangs the success of the engine.

No small Diesel engines employ the "single rail," or accumulator, fuel pump. All have a separate pump cylinder for each engine cylinder.

Fuel pumps may be either "single unit pumps" or "multiple unit" pumps. The latter type may have a combined pump and injection nozzle.

(*a*) The "*single unit fuel pump*" consists of a block in which are installed all the pump cylinders, the suction valves, the by-pass valves, and the discharge valves. The fuel pump pistons are actuated usually by a camshaft driven by a gear, small throw crank, eccentric, or even a chain. It is a complete integral pump unit and regulating mechanism. The type usually employs a separate camshaft and drive. (See Figure 1.)

(*b*) The *multiple unit type* has a separate pump for each cylinder. It is located at the cylinder which it supplies and is operated like the exhaust and intake valves by a cam on the large camshaft.

(*c*) *The combined pump and nozzle.* To avoid long fuel leads to the valves which cause column action, after-dripping, and carbonization, a combined pump has been used in which the pump and closed automatic nozzle are integral. The needle valve of the nozzle acts as a spring-loaded automatic discharge valve.

In summarizing the types of fuel pump construction, the usual form is the "single unit." Its merits are:

(*a*) Easier adjustment and repair than multiple.

(*b*) Simpler to install and time than multiple unit.

(*c*) More uniform fuel measurement.

(*d*) More adaptable to specialty manufacturing.

(*e*) Less complicated, more compact an accessory than many units all "hung on" the engine.

Its demerits consist of:

(*a*) Longer unequal fuel leads which tend to cause inertia in lines or column action, unbalance of cylinder fuel supply, and carbonization of the injection valve.

(*b*) A more complicated drive consisting of extra camshaft and drive gear.

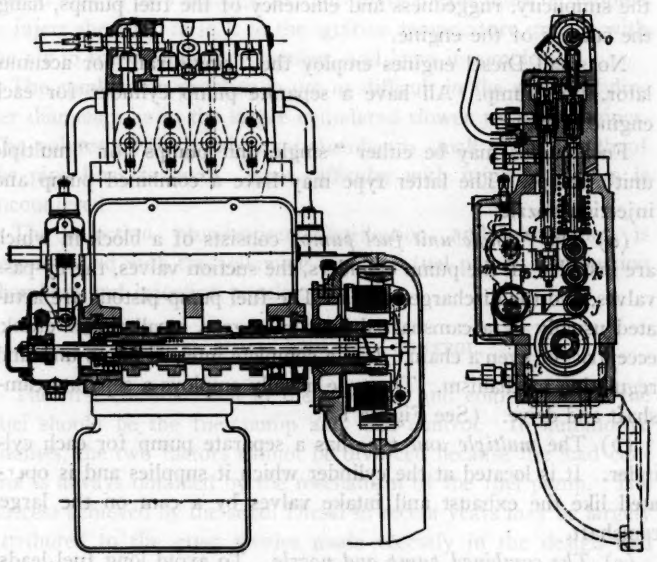


FIG. 1.—SECTION OF COMPLETE BYPASS VALVE, SINGLE UNIT FUEL PUMP AND DRIVE.

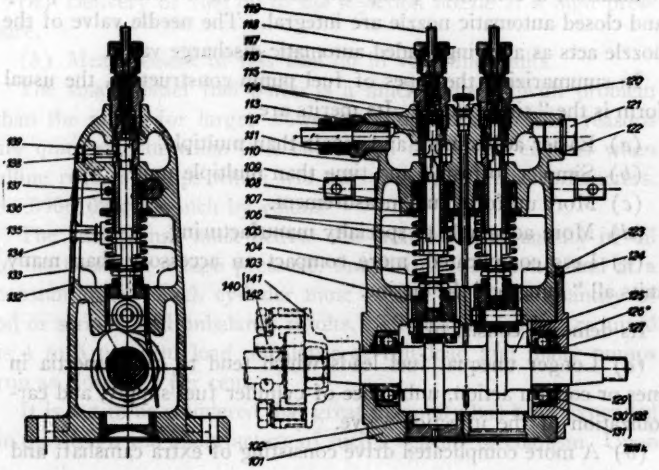


FIG. 2.—ROTATING PLUNGER TYPE OF SINGLE UNIT FUEL PUMPS.

The trend in larger engines is toward the "multiple unit pump" because of the short length fuel leads and the uniform cylinder loading. Much more space is available on the camshaft and cylinder block of the large engine.

On the small Diesel, however, the "single unit" is much more common, partly because the fuel leads are all quite short, there is very little room on the cylinder block and camshaft, and the installation is simpler. The controlling reason for the single unit pump on small Diesel, however, is the fact that fuel pump manufacture is so delicate that it has been taken over by the specialty or accessory manufacturer, who has standardized fuel pumps like spark plugs, or carburetors, and who prefers to sell a complete pump unit in a single block.

Robert Bosch now manufactures "single unit" fuel pumps up to about 70 H.P. per cylinder. Bosch has undertaken it not only because he has dipped into the future, and he foresees the advent of the automotive Diesel, but also because he realized that the manufacture of solid injection fuel pumps properly involves a degree of tolerance seldom demanded commercially (.00001), special equipment, highly skilled men, and a capital outlay not possible to the engine manufacturer building pumps only for his own needs.

LOAD CONTROL.

The small Diesel fuel pump must measure out the fuel to each cylinder equally and also vary the amount of fuel from no load to full load. Variable loading need not be stressed as much in the motorboat engine, which is always running nearly full load, but it is necessary to idle when the clutch is thrown out and it is desirable to slow down for maneuvering. Great flexibility is not necessary, but moderate flexibility is desirable.

Fuel pump load control is obtained by the following devices:

(*a*) Variable stroke of pump piston.

(1) Conical cam plunger drive.

(2) Fulcrum lever between cam and plunger tappet.

(3) Sliding wedge between cam and plunger tappet.

(*b*) Constant stroke and variable bypass.

(1) Control by mechanically operated spill or bypass valve.

(2) Control by by-passing through port and rotating helical piston port.

VARIABLE STROKE FUEL PUMPS.

(*a*) Conical Cam Control (Variable throw cam). The stroke of the pump may be changed by the use of a conical cam, the lift of which would consequently vary along its axes. It is moved longitudinally under the cam roller by the throttle lever and this movement varies the stroke of the pump, hence varies the fuel delivered to each cylinder, and consequently changing the load.

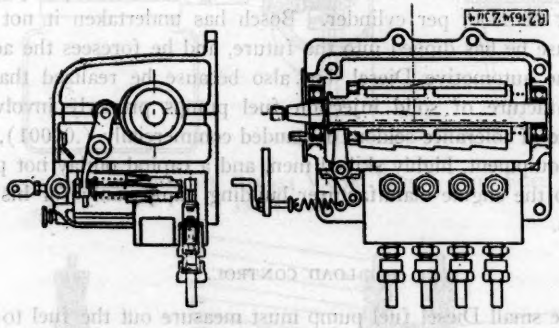


FIG. 3.—CROSS SECTION OF FUEL PUMP, VARIABLE STROKE CONICAL CAM CONTROL.

(*b*) Fulcrum Lever Control (Constant Throw Cam). If between the cam and plunger tappet, a lever is interposed such as "h" in sketch below, and the tappet roller "V_2" be moved away or toward the fulcrum "O" by the throttle shaft "n" and links "t," the stroke of the pump plunger will vary almost directly as the distance of "V_2" from the fulcrum "O."

(*c*) Sliding Wedge Control (Constant Throw Cam). If in the sketch (Figure 4) the arm "h" be made wedge-shaped, the plunger tappet roller "V_2" be fixed, and the fulcrum "O" be

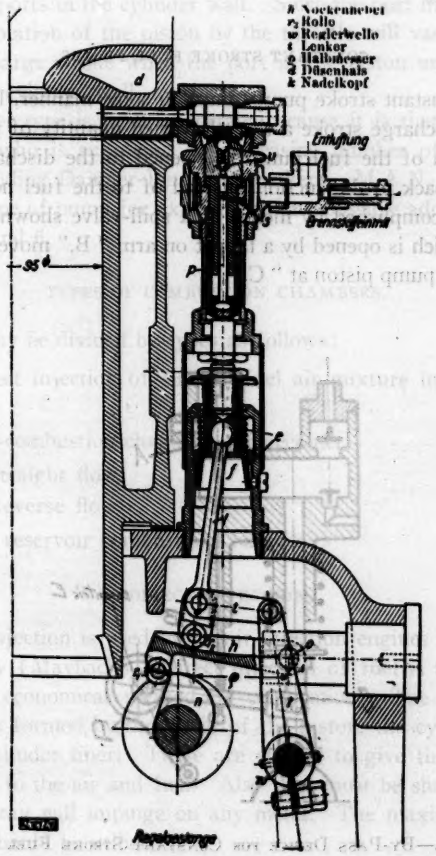


FIG. 4.—FUEL PUMP WITH FULCRUM LEVER CONTROL.

moved right or left by the throttle lever, the cam roller clearance, and hence the pump stroke, would vary with the position of the wedge " h."

CONSTANT STROKE FUEL PUMPS.

(*a*) Constant stroke pumps must in some manner, by-pass during the discharge stroke a throttle-varied quantity of the fuel. If the suction of the fuel pump be opened to the discharge, the oil will flow back to the suction, instead of to the fuel nozzle. This may be accomplished by means of a spill-valve shown at " A" on sketch, which is opened by a tappet on arm " B," moved with relation to the pump piston at " C."

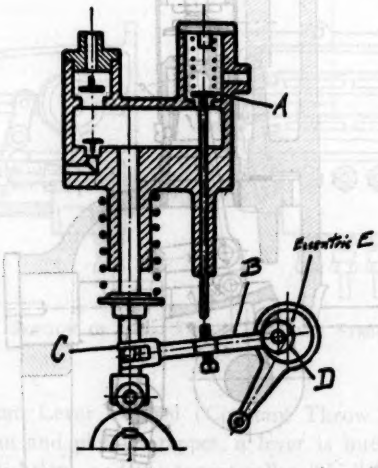


FIG. 5.—BY-PASS DEVICE FOR CONSTANT STROKE FUEL PUMP.

If the throttle rotate the eccentric " E," the clearance between the spill valve spindle and its tappet will increase or decrease, thereby changing the time of opening of the spill valve with relation to the pump discharge stroke. Clearance increase delays the opening of the bypass and increases fuel delivery and load.

(*b*) Another method of by-passing the fuel is by means of a by-pass port in the fuel pump piston between the discharge and the suction ports in the cylinder wall. Since the port in the piston is helical, rotation of the piston by the throttle will vary the time in the discharge stroke when the port in the piston uncovers the port in the cylinder wall.

This latter type is most important because it is that advocated by Bosch, who is supplying an increasing number of manufacturers, including Daimler-Benz and Junkers. M.A.N. is building a similar type of pump for experiment, and possible adoption. See Figures 2 and 6.

TYPES OF COMBUSTION CHAMBERS.

These may be divided by types as follows:

(*a*) Direct injection of fuel or fuel air mixture into combustion space.

(*b*) Pre-combustion chamber or forecomer.

(1) Straight flow.

(2) Reverse flow.

(*c*) Air reservoir combustion chamber.

DIRECT INJECTION.

Direct injection is used by all air injection engines with needle valve spray (Maybach). Direct injection of fuel is the simplest and most economical method of combustion. The combustion chamber is formed by the metal of the piston, the cylinder head, and the cylinder liner. These are shaped to give the maximum turbulence to the air and fuel. Also they must be shaped so that no fuel spray will impinge on any metal. The maximum length of spray should be utilized.

Direct injection must depend entirely for breaking up the oil upon the hydraulic pressure of the pump, and upon the mechanical atomization of the oil in the specially shaped nozzles. Pump pressures as high as 3000 to 5000 pounds are normal. Junkers uses 10,000 pounds pressure.

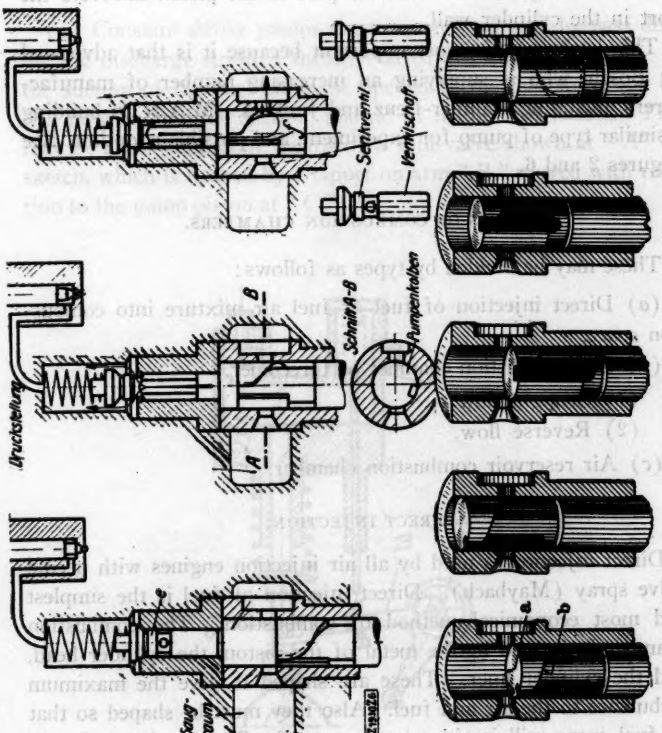


FIG. 6.—SKETCH ILLUSTRATING PRINCIPLE OF ROTATING PLUNGER CONSTANT STROKE FUEL PUMP.

Direct injection at very high pressures causes high maximum pressures. The combustion rise on these engines is seldom under 150 pounds, and the Junkers engine runs a maximum pressure of 1050 pounds.

In spite of its faults, many of the leading manufacturers use direct injection. See Table. The trend appears to be in that direction. The small cylinder sizes, the metallurgical advances of recent years, and the constantly improving design of injectors, pumps and chambers seem to have solved the difficulties.

Direct solid injection has the following faults:

(*a*) High fuel oil pressures induce pump, packing, and pipe line troubles.

(*b*) High maximum pressures cause material failures, mechanical shocks, and vibration.

It has the following advantages:

(*a*) It is at least 10 per cent more efficient than the forecomer combustion.

(*b*) The engine starts more easily.

(*c*) Greater simplicity and ruggedness of head.

FIG. 6.—SKETCH ILLUSTRATING PRINCIPLE OF ROTATING PLUNGER CONSTANT STROKE FUEL PUMP.

SOLID INJECTION WITH A FORECOMER.

Some manufacturers prefer to "dodge" the difficulties met by the direct injection method by using a pre-ignition chamber at the expense of economy and simplicity.

The forecomer is an auxiliary combustion chamber or antechamber with a volume equal to about 20 per cent of the compression space. It is connected to the main combustion chamber by one or more orifices.

The fuel oil is injected into this space during compression several degrees before injection would normally occur. The oil unites with the air rushing into the chamber through the orifices.

Injection in the antechamber occurs when the pressure has risen to give the charge the ignition temperature. The fuel charge is partially burned.

The consequent combustion causes the forecomer pressure to rise above the main chamber pressure. Oil, air, and burning gases, impelled by the pressure difference, then rush into the

main chamber and the unburned oil, now completely atomized, is carried into the air of the combustion chamber at high velocity and great turbulence. Excellent combustion results because the oil is undoubtedly pre-heated.

ADVANTAGES.

(*a*) The pre-combustion chamber permits the burning of very low grade fuels.

(*b*) Due to the fact that the oil *should not be* completely atomized by the injection nozzle, the pump pressures are only about 1200 pounds.

(*c*) It is, therefore, also possible to employ far cruder injection nozzles with larger holes which will not clog or require fine fuel filters.

(*d*) The maximum pressures and compression pressures may be lower.

DISADVANTAGES.

(*a*) On the other hand, forecomers, employing hot gases for atomization, are at least 10 per cent less efficient than direct solid injection.

(*b*) Difficulty is also experienced in cooling the preignition chamber because of the very high temperatures developed.

(*c*) They are much harder to start and most types employ a glow-plug for starting cold.

(*d*) They are also slightly heavier, larger, and more complicated than direct solid injection.

STRAIGHT FLOW CHAMBERS.

In Figures 7 and 8, the oil is sprayed through the nozzle "d" into the chamber "V." Through the orifices "K," the air from compression space rushes into the chamber and mixes with the oil spray. As soon as the pressure and temperature is sufficiently high, combustion occurs and the oil and air rush at high velocity into the main chamber "C."

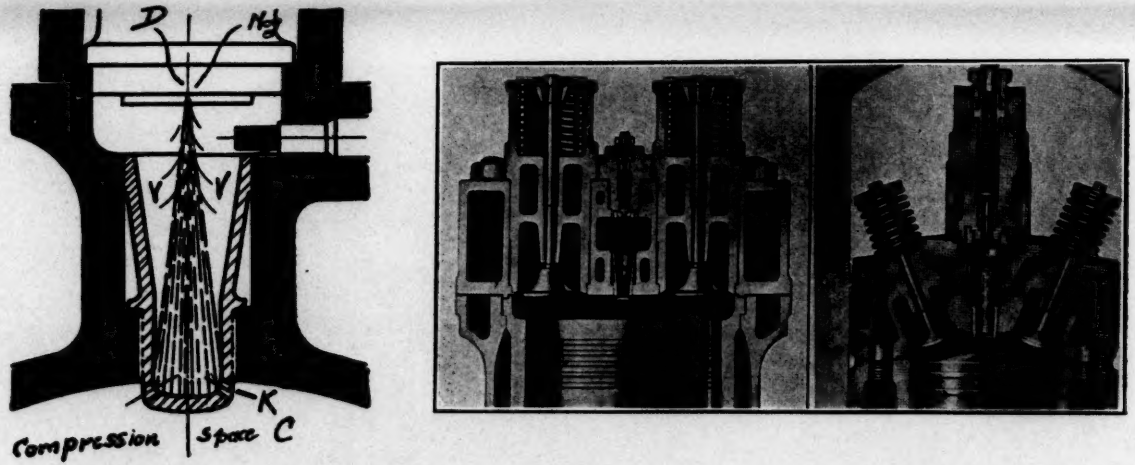


FIG. 7.—DIRECT FLOW PRE-COMBUSTION CHAMBERS.

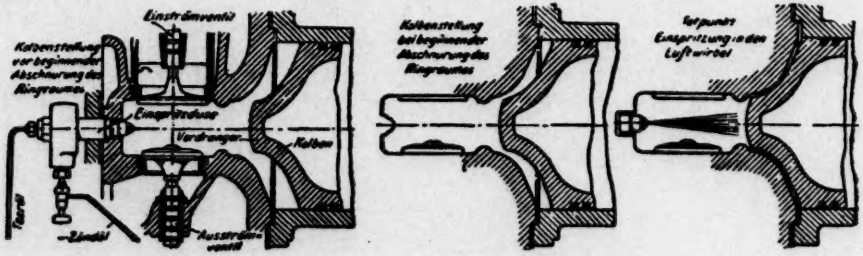


FIG. 8.—DEUTZ COMBUSTION CHAMBER.

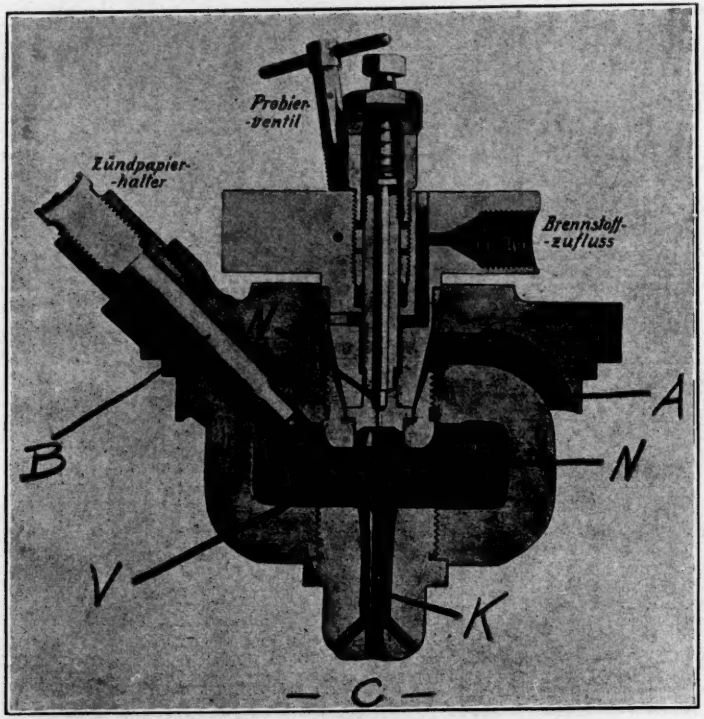


FIG. 9.—REVERSE FLOW PRE-COMBUSTION CHAMBER.

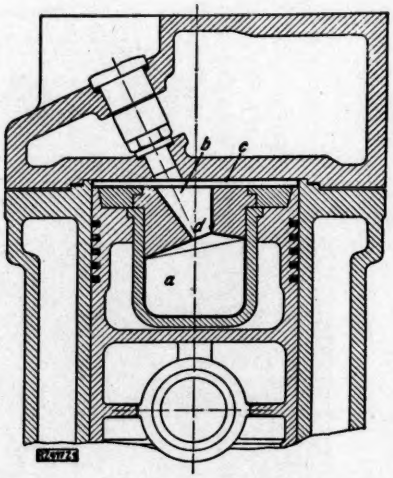


FIG. 10.—DIAGRAM OF AERO AIR RESERVOIR—PISTON TYPE.

REVERSE FLOW FORECOMERS.

In Fig. 9 the fuel spray through nozzle "N" impinges on the walls of the orifice "K." The injection begins during the compression stroke while air from main chamber "C" is rushing into chamber "V."

The air rushing through orifices "K" tears the oil from the walls of the orifices and carries it back into chamber "V" in a finely divided state.

As soon as the pressure and temperature rises sufficiently these oil globules ignite and the pressure rises. The ensuing action is similar to the direct flow type.

AIR RESERVOIR COMBUSTION CHAMBER. FIG. 10.

The Lang air reservoir patents are owned by the Acro Company, Switzerland, who have granted a German license to Robert Bosch. Bosch has sub-licensed manufacturers such as Orenstein Koppel, Reform, and others. The trade name, Acro Bosch, is copyrighted in Germany.

A diverging nozzle "B" connects the air reservoir in the piston "A" with the combustion chamber "C." The clearance volume above piston is as small as safety permits, and all the compression air is forced into the air reservoir "A" and the nozzle "B."

As the oil is sprayed from the nozzle, it first meets the hot compressed air in the orifice "B." As the piston moves down the pressure in "C" falls and the air in "A" rushes through "B" to meet the vaporized oil.

In the second design, Figure 11, the air reservoir has been placed in the head in order to avoid the heavy piston and reciprocating weights necessitated by the type using an air reservoir and to provide for the better cooling of the air chamber. Its operation is self-evident.

The air reservoir combustion chamber is, in effect, merely a forecomer with a volume of 95 per cent of the compression space instead of 20 per cent. Its patentability and novelty rests on this fact, on the use of an air chamber in the piston, and on the fact that entire combustion occurs during working stroke.

FIG. 9.—REVERSE FLOW PRE-COMBUSTION CHAMBER.

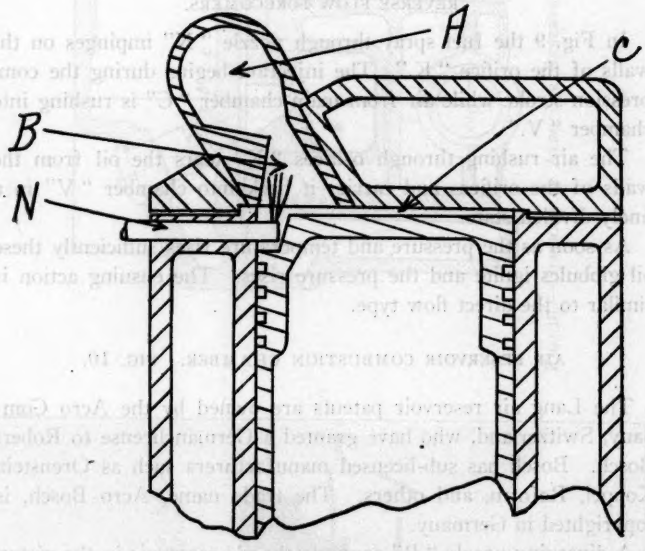


FIG. 11.—DIAGRAM OF ACRO AIR RESERVOIR—HEAD TYPE.

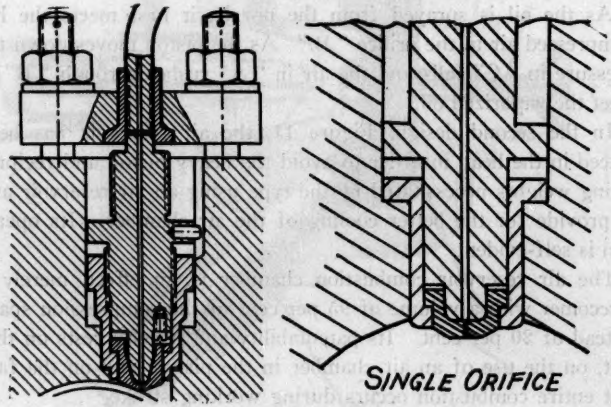


FIG. 12.—DIAGRAMMATIC SKETCHES OF OPEN ORIFICES.

The International Harvester Company is believed to have an interest in the American Acro rights.

TYPES OF INJECTION NOZZLES.

(*a*) Injection of fuel by compressed air.
(*b*) Injection in solid spray.

COMPRESSED AIR INJECTION.

The trend in small Diesel design is toward solid injection. There is only one advocate of air injection, namely, Maybach who builds only one type of Diesel engine, a 150 H.P., 6 cylinder, 1300 R.P.M. engine.

The air compressor engine fails to meet the requirements of simplicity and ruggedness. Its first cost is higher. It is harder to start because of the refrigeration of the expanded air fuel mixture. Due to refrigeration, it requires a higher compression with consequent greater weight, more severe vibration, and possibly shorter life.

The requirements of injection, mixture, and combustion are excellently met, and its maximum pressure due to combustion pressure rise is low.

The Maybach injection valve is a cam operated, constant stroke, hollow needle valve with load controlled by an inner solid tapered needle.

SOLID INJECTION.

The following manufacturers are building automobile, rail car and motorboat engines with solid injection.

M.A.N.	Krupp Germaniawerft
Benz-Daimler	Koerting
Deutz	Linke-Hoffman
Fiat (Italy)	Burmeister-Wain (Denmark)
Beardmore (England)	Junkers

FUEL INJECTION NOZZLES.

The following types of injectors are used:—

(*a*) Open nozzles.

(*b*) Closed nozzles.

OPEN NOZZLES.

The tendency is toward the open nozzle because it is simpler and has fewer moving parts. It requires, however, very high grade materials and excellent workmanship because it is sometimes necessary to drill .005 inch holes with the greatest accuracy. The smallest irregularity in drilling or angularity of the holes causes bad combustion. However, special equipment and skill enables these to be manufactured cheaply, and one firm supplies them like spark plugs.

Another serious objection to the open type of injector is the after dripping of fuel due to:

(*a*) The inertia of the column of oil in the fuel line to the valve.

(*b*) The contraction in the expanded fuel line due to the release of pressure during the expansion stroke.

(*c*) The similar expansion of the entrained oil upon release of pressure during expansion stroke especially when oil contains minute air bubbles.

(*d*) Setting up of column and pipe vibrations in fuel line.

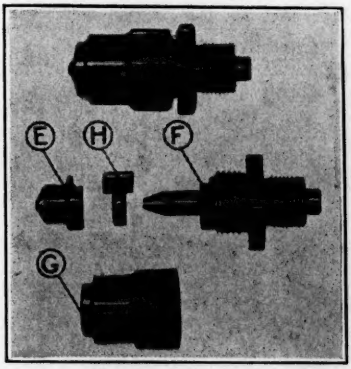
After dripping causes dirty combustion, carbonization around nozzle holes, and load unbalance. The means adopted to obviate after dripping are:

(*a*) Reduction of velocity and acceleration in the fuel line by increasing the size and cross section of fuel line to nozzle.

(*b*) Increasing the wall thickness and rigidity of the fuel to prevent expansion and contraction of pipe.

(*c*) Eliminating air in fuel by the adoption of very rigid standards in packing or lapping in the fuel pump plunger.

(*d*) Reducing vibrations in the pipe and the oil column by making the fuel line of such a length and period of vibration that no



The Assembled Nozzle Is at the Top. Below Are the Parts: *E*, Nozzle Tip; *F*, Body; *G*, Nut; *H*, Collar.

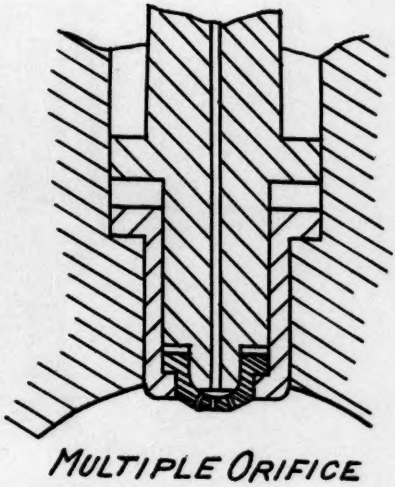


FIG. 13.—M. A. N. OPEN TYPE INJECTION NOZZLE.

FIG. 14.—BOSCH OPEN TYPE INJECTION NOZZLE.

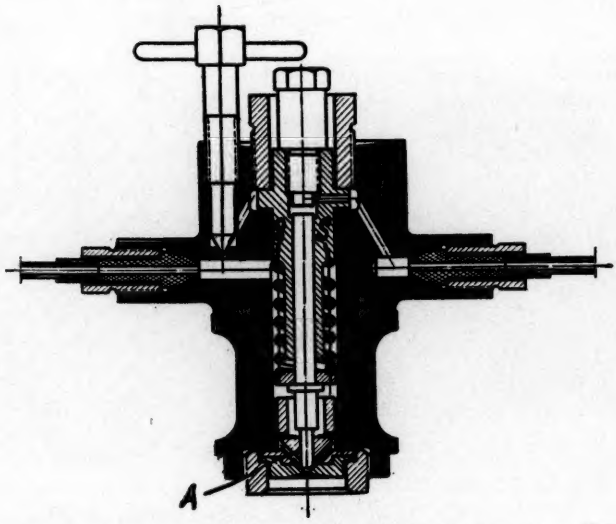


FIG. 15.—DIAGRAMMATIC SKETCH OF TYPICAL CLOSED AUTOMATIC NOZZLE.

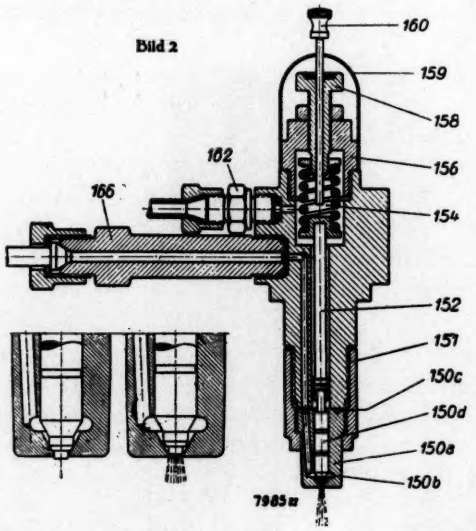


FIG. 16.—SKETCH OF BOSCH TYPE CLOSED AUTOMATIC INJECTION NOZZLE.

vibrations are induced in synchronism with fuel pump delivery or cylinder cycle.

(*e*) Placing the pump as close as possible to the fuel valve so that the inertia of oil in pipe is reduced.

Another objection to the open nozzle is offered by the difficulty of filtering the fuel oil and cleaning it of all foreign matter larger than .005 inch. The open nozzle therefore requires a very refined filter and these filters are a source of trouble. However, there are 3 or 4 types now on the market which appear to have solved the difficulty.

Notwithstanding the objections cited, the trend appears to be toward the open nozzle.

CLOSED NOZZLE.

This type of nozzle is still adhered to by several manufacturers because less trouble is encountered with after drip. A schematic drawing of the closed nozzle is shown in Figure 15.

The closed automatic nozzle consists of a needle valve "A" closing an orifice into combustion chamber. This needle is held on its seat by spring "S." As soon as the fuel pump commences the injection stroke, the pressure in the passage marked "fuel oil intake" acts on the piston "P" which opens the valve against the tension of the spring "S." The valve closes as soon as the pressure drops. The amount of the pressure drop required to close the valve, and therefore the total valve opening, depends upon the tension of the spring "S." This tension may be adjusted by the nut on the spring bushing.

No article on European automotive Diesel spray nozzles is complete without reference to Bosch, and Figure 16 is a drawing of this nozzle which are sold to many manufacturers and which may be purchased here in all garages. Its operation is similar to that just described.

Another type of closed automatic nozzle is shown in Figure 17. This type employs ball valves and the valve is the mushroom type opening into the cylinder and against the cylinder pressure instead of with the pressure.

DESIGN FACTORS WHICH HAVE BEEN INFLUENCED BY AUTOMOTIVE AND AIRCRAFT CONSTRUCTION.

The design factors of injection and combustion which influenced the cyclic characteristics of the engine have been discussed. These design factors are purely Diesel and no parallel can be drawn between these elements and the construction of a gasoline engine.

The dissimilarity between the Diesel engine and the gasoline engine ends here. From this point on in this discussion it will be found that the gasoline engine and the Diesel engine are almost similar. The old types of Diesel engines were all developments of the steam engine or of the horizontal gas engine, and for thirty years the entire design of the Diesel engine was influenced by these old designs. The last 10 years, however, the Diesel has entered the automotive field and the entire design is now either basically an automotive gasoline engine design or in one case at least the Diesel engine shows the influence of the aeroplane engine.

FRAMING.

The framing of the Diesel engine is very similar to that of the gas engine with the following exceptions:

(*a*) A more extended use of aluminum in the crank case and housing.

(*b*) In some cases, the use of small nickel steel tie rod, in addition to the box frame structure of the gasoline engine.

The following is quoted from a paper by Dr. Wilhelm Riehm of the M.A.N. Company:

"The question may be asked whether these higher stresses are such that, because of the more rugged construction required, the weight of a light engine operated on the Diesel principle would be appreciably higher than that of the carburetor engine. If this question is to be answered first respecting the working parts this must be done on the basis of the maximum pressures. In the case of carburetor operation, in which the explosion pressure cannot be controlled with the same certainty as in Diesel engines, the maximum pressures have been up to 28 atmospheres or more in

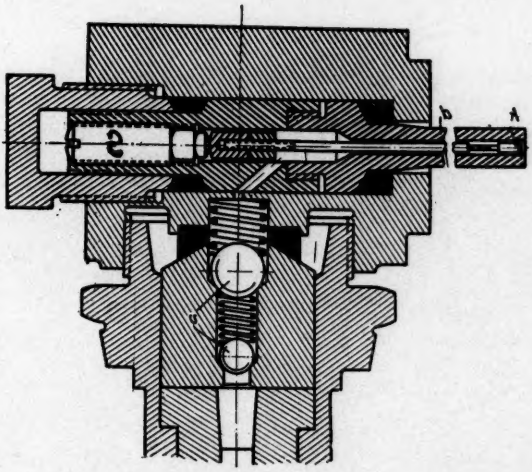


FIG. 17.—SKETCH OF MUSHROOM TYPE CLOSED AUTOMATIC INJECTION NOZZLE.

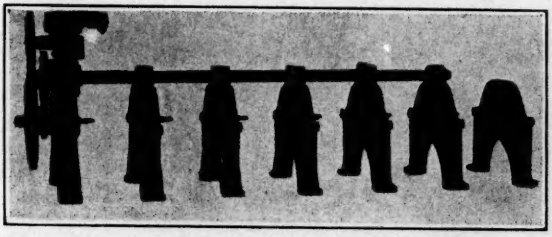


FIG. 18.—A NOVEL LUBRICATING OIL PUMP.

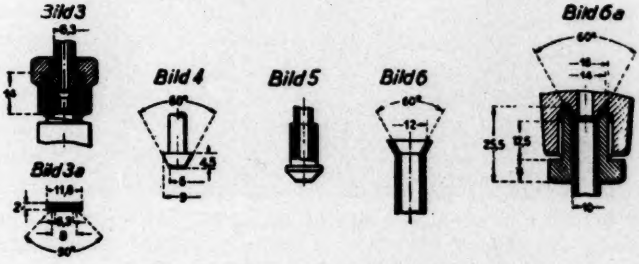
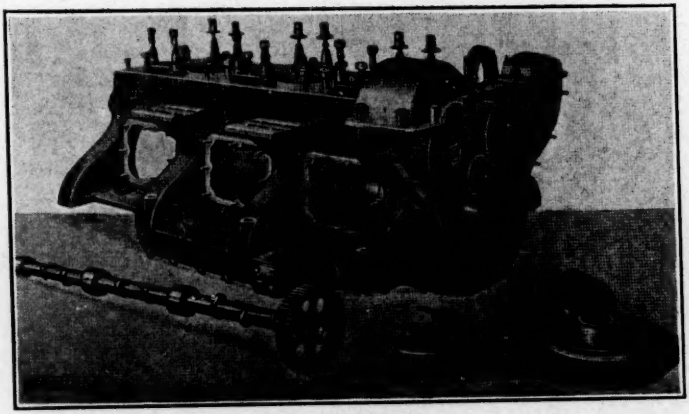


FIG. 19.—MAKING FUEL LINES TIGHT.



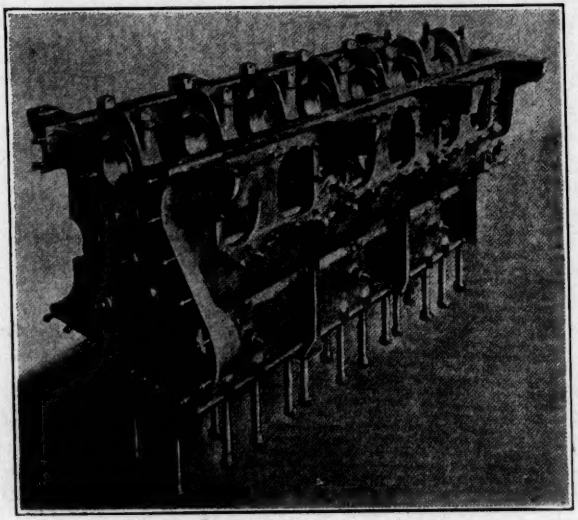


FIG. 20.—ALUMINUM CRANK CASE.

recent designs. When using the Diesel principle, a combustion pressure of 42 atmospheres may generally be assumed for high speed engines.

"The diameter of the connecting-rod is governed by the maximum cylinder pressure, increasing with the 4th root of the pressure. Hence, raising the pressure 1.5 times increases the diameter of the connecting-rod 10 per cent and its weight about 21 per cent, presupposing a connecting-rod of circular cross-section. The dimensions of the crankshaft are based upon the maximum tangential pressure at a crank angle of 30 to 35 degrees, and the crankshaft must be 4 to 5 per cent heavier than in the carbureter engine.

"Because of casting and manufacturing considerations, the wall thickness of cylinders, cylinder-heads, pistons and crankcase generally are so heavy that they will not need to be made heavier for the higher ignition pressures. The fuel-injecting device provided instead of the carbureter also can be very light, so that the Diesel engine will weight only 10 to 15 per cent more than a carbureter engine of the same cylinder displacement.

"Since, in the vehicle engine, the weight is always related to the attainable power, one must investigate also how the power will compare in the carbureter engine and the Diesel engine for the same cylinder displacement. The power depends on the attainable mean effective pressure and the highest practicable speed. Carbureter engines usually attain mean effective pressures between 6 and 7 kilograms per square centimeter (85 to 100 pounds per square inch). In Diesel engines that have been in satisfactory operation on cars for years, mean effective pressures up to 6 atmospheres (88.2 pounds per square inch) are obtained normally with good combustion. At lower speeds mean effective pressures greater than 7 atmospheres (102.9 pounds per square inch) can be attained for short periods. On the whole it can be said that the Diesel engine will hardly fall below the carbureter engine in driving torque.

"Whether greater limitations are imposed upon the speed range of the Diesel engine cannot be decided now, but, with the same normal speed, the lowest speed at which the engine will ignite regularly probably will be the same as for the gasoline engine.

Also the upper limit of around 2000 R.P.M. required for truck engines, probably can be reached without difficulty. From this it follows that, with the same cylinder volume, approximately the same power will be given by the Diesel engine as by the carbureter engine and the weight will be from 10 to 15 per cent greater for the Diesel engine, which would not prevent the use of this engine in vehicles."

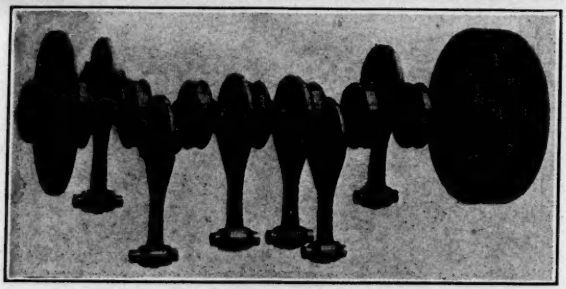
PISTONS.

In order to reduce the heavy strain on the piston caused by the high compression and combustion pressures, and to use the higher heat conductivity of aluminum, the Diesel engine uses more extensively aluminum pistons. These aluminum pistons are usually made of a special aluminum alloy in which surface hardness is stressed, and these pistons are bought from specialty manufacturers such as Schmidt in Neckarsuhn, who deliver pistons completely finished up to about 200 millimeters diameter for less than $5.00. In Europe there are few, if any, users of the invar ring piston construction.

CONNECTING RODS.

The connecting rod must withstand at least 1½ times the pressure obtaining in a gas engine, and hence its cross section must be increased about 10 per cent and its weight about 21 per cent, according to Dr. Riehm. Connecting rods are made both with circular cross section and with the I-type section, and they may be made of high grade nickel steel or in one case at least, they are made of duraluminum in order to further reduce the reciprocating weights. In most cases the connecting rods are drop forgings, and in general connecting rod practice follows the practice in gasoline engines with the exceptions noted.

The crank shafts are generally the same as gas engine crank shafts with the exception that they must be 4 to 5 per cent heavier again to withstand the higher pressures of the Diesel engine. Crank shafts are usually forgings purchased from specialty manufacturers such as Krupp at Essen, and many others. In one case, a built up crank shaft is used and extreme refinement in crank



THE CRANKSHAFT ASSEMBLY.

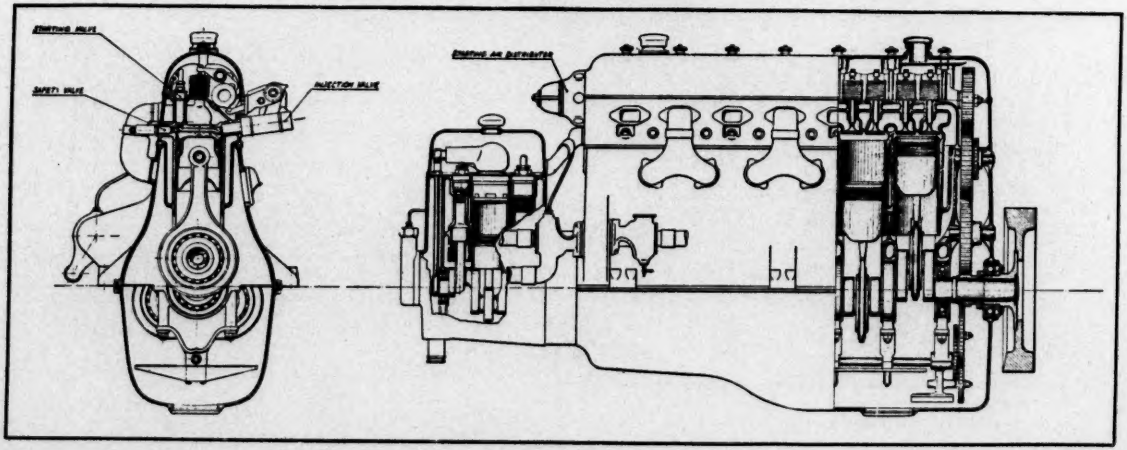


FIG. 21.—MAYBACH ROLLER BEARING CONSTRUCTION.

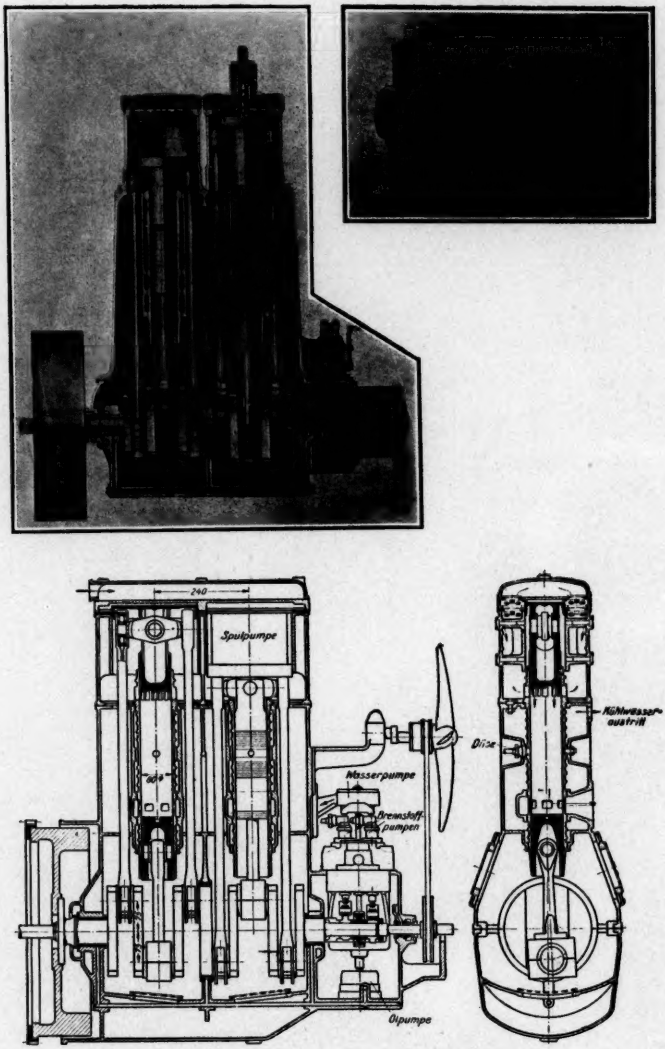


FIG. 22.—JUNKERS OPPOSED PISTON ENGINE.

shaft balancing in the automotive Diesel engine is resorted to because of the tendency of the Diesel engine to vibrate more than the gas engine.

VALVES.

The valves in design usually closely follow the design of the gas engine valve. Again to withstand the greater shock due to pressure, they are made with slightly heavier stems and the valve metal itself is often chrome vanadium steel.

CYLINDER DESIGN.

The cylinders for the Diesel motor boat engine are usually cast en bloc but are often cast in pairs. In the larger sizes above 100 H.P., two manufacturers cast the cylinders singly and bolt them together with body-bound bolts. The cylinder casings are sometimes made of aluminum and the liners removable. In this case it is necessary to use an aluminum alloy which is corrosion resisting and one manufacturer supplies this aluminum under the name of KS-Seewasser. This aluminum has been experimented upon and endorsed by the Italian Navy at their laboratory in Spezia. Cylinder liners if detachable, are made of a special cast iron alloy which is called "cylinder iron" and the specifications of which are as varied as the manufacturers. A hard, coarse grained gray iron, in some cases of more or less pearlitic structure, is usually specified.

In conclusion it may be pointed out again that all of the above features of design are not dissimilar from gas automotive and air craft engine practice.

MISCELLANEOUS.

With the open type of fuel nozzle, it is necessary to stress the fine filtering of the fuel oil. Several types are on the market one of which, the Autoclean, is used by M.A.N. and others, and consists of parallel discs, fixed and rotating, which are similar to a radio condenser. This filter may be cleaned by rotating a small castle nut on the filter cover. Krupp Germania Werft employ a synthetic stone filter which has given excellent results on an auto-truck during 12,000 miles of operation.

Many other types are on the market, most of which employ clearance between ground threads or fine passages. The Hesselmann strainer uses this principle. In general it may be said that several types of fuel filters have met the rigid demands of Diesel open nozzle type of fuel injection.

FUELS.

The manufacturers of light high speed automotive Diesels give the following acceptable fuels: "cheap tax-free Diesel engine fuels such as Mazout, petroleum, solar oil, gas oil, Texas oil of a specific gravity of .83 to .88, lignite gas oil, and yellow and brown paraffine oil." Another specifies Diesel oil such as crude oil, gas oil, petroleum, lignite oil, tar oil, paraffine oil, solar oil, shale oil, etc. It has been noted that some manufacturers specify "any cheap Diesel oil except tar oil."

FUEL ECONOMY.

The small high speed Diesel has less efficiency than the larger Diesel and a good conservative average fuel consumption for the automotive Diesel is 200 grams per B.H.P. per hour or .45 to .50 pounds of oil per brake horsepower hour.

LUBRICATION.

Unlike the large slow running engines, the lubrication of the small high speed Diesel, four cycle, single acting engine, is not difficult. All types lubricate the cylinder walls by splash and the chief difficulty with lubrication of the walls is to prevent overlubrication. Special types of piston rings are sometimes employed by those manufacturers whose designs are subject to this difficulty.

BEARINGS.

In order to withstand the high compression pressures and shocks, special bearing construction is used by most manufacturers and in order to prevent the crank shafts from having excessive length, due to long heavy bearings to withstand the combustion pressures, at least two manufacturers use roller bearings throughout thereby shortening the engine considerably and reducing the friction horsepower.

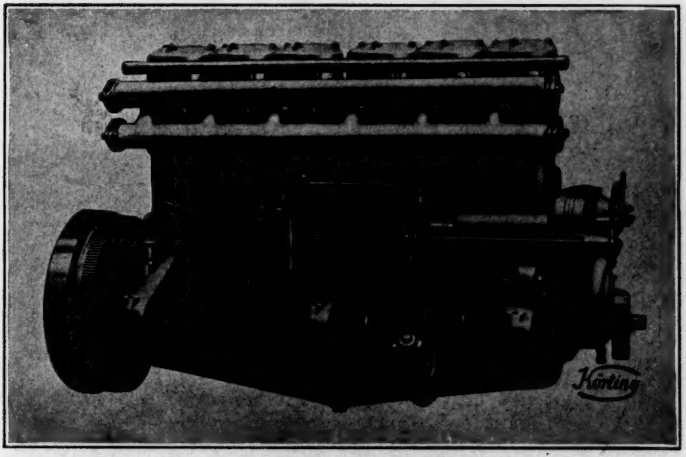


FIG. 23.—KOERTING HIGH SPEED ENGINE.

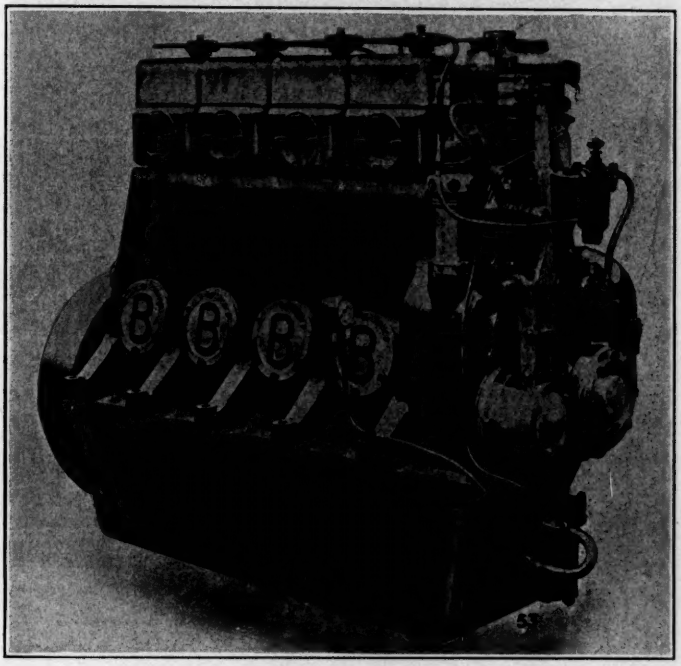


FIG. 24.—BEARDMORE.

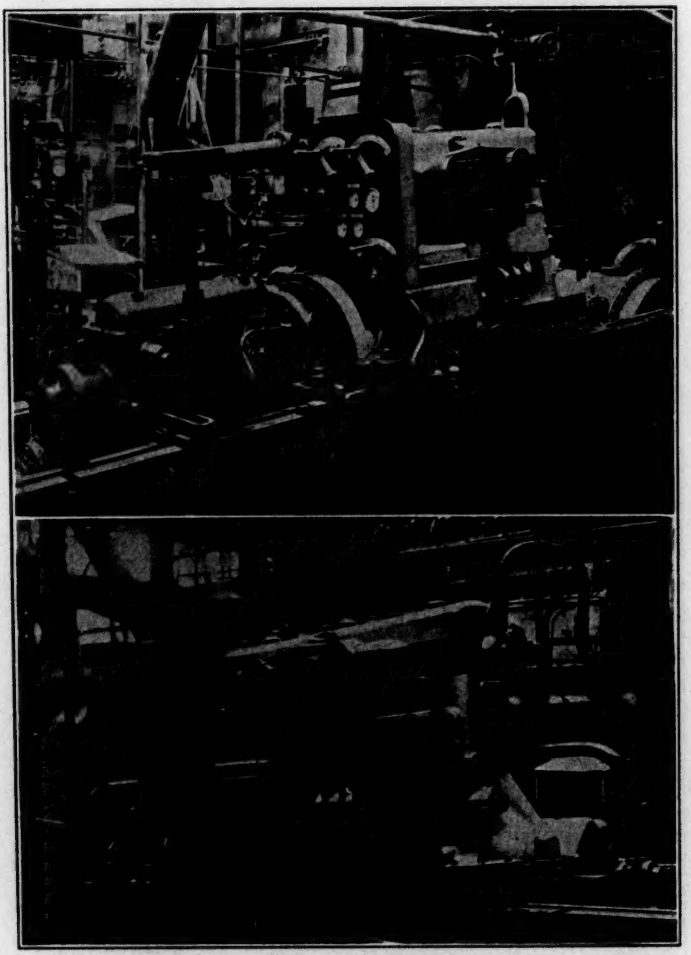


FIG. 25.—KRUPP GERMANIAWERFT.

STARTING.

The Diesel engine in general starts with more difficulty than the automotive Diesel. This is particularly true of the forecomer or antechamber type of combustion chamber, and in this type it is always necessary to fit the head with a glow plug or glow paper cartridge for cold weather starting.

Many of the automotive Diesel engines of the smaller sizes start electrically, similar to the gasoline automotive engine but uses a compression release which relieves the compression until the engine is spinning.

Another type of small Diesel uses an air motor with the Bendix pinion release. Some manufacturers in the motor boats use a small hand compressor like an automobile tire pump and the compressed air thus obtained is used, either to run an air starting motor or it is used with an air starting valve on the cylinder head.

In general the smaller types of high speed motor boat forecomer Diesels may be started electrically, although many must employ some sort of starting auxiliary. The small direct injection engine may be started just like the automobile gasoline engine. The larger engines from 100 to 200 H.P. must be started by air.

CLUTCHES.

The clutching of the Diesel motor boat engine presents the same problems as the gasoline motor boat engine clutch. Krupp is designing a new clutch which extends only 18 inches from the engine frame. In general it is probably necessary to build the Diesel clutch more rugged than the gasoline motor clutch, due to its high starting torque, pick up, and the shocks of high compression.

This article has sketched the past and present of the entrance of the Diesel into competition with the gasoline mixture or Otto engine.

Future development is largely dependent upon overcoming or circumventing those factors which limit the speed and cylinder power output. If these limits be raised, the factors of safety, ruggedness, weight reduction, and increased life will follow.

Three American firms, great truck manufacturers, have combed Germany recently and acquired German licenses to build automo-

tive Diesels. The work of Buda-M.A.N. in America is well known. Lilloise, in France, is manufacturing Junkers Diesel trucks.

Nearly all builders of Diesels in Europe are already selling or are running experimentally high-speed, light weight, automotive engines.

The aircraft engine developments of Junkers, Packard, Beardmore, and M.A.N. strongly demonstrate the faith of great engineering firms in the future of the high speed Diesel.

This discussion has shown that a dozen manufacturers have placed upon the market an engine which will undoubtedly make motor boats safer for their passengers and more economical to operate.

The Diesel has gone on to enter a service far more exacting than that met in Navy Motor Boats.

Small high speed Diesel engines for motor boats. Brake horsepower less than 200. Speed over 1000 r/min.

Manufacturers. Name.	Number.	B.H.P.	R.P.M.	No. of cylinders	Bore ins.	Stroke ins.	Displacement Cubic Inches	Displacement Litres	Valve Arrangement	Fuel Injection	Type of Combustion chamber.	Type of fuel pump.	Remarks.
I M.A.N. (Buda) a	MVH 11/16	45	900 to 1000	4	4.7	7.1	22		Solid	Open	Direct injection combustion chamber.	Single unit constant stroke and by pass valve.	In production - Several hundred are running in trucks various manufacturers. Admiralty is using this type in motorboats. Motorboats B.H.P. 200-12-40 Immersion - 800' - 45
	MGH 11/16	68	900 to 1100	6	4.7	7.1	22						
	MHV 14/22	90	800	4	6.5	8.7	20						
	MGH 14/22	135	800	6	6.5	8.7	20						
	MGH 14/22	180	800	8	6.5	8.7	18						
II Junkers. 2-Cycle. a	SA9	45	1100	2	3.2	4.8		22	Solid	Open	Direct injection.	Single unit Constant stroke reversing plunger.	Reversed valve - Brake given at 1100 r/m and 45 B.H.P. Reversing valve 10 per cent. Starting gear fitted for experiment.
	SA12	70	1400	3	3.2	9.5		21.3					
III Daimler-Benz.		70	1300	6	4.1	6.5	19	21	Solid	Closed	Precombustion chamber.	Same	In production for oil boat by Daimler Benz, Stuttgart.
IV Maybach a	G4A	150	1300	6	5.5	7.1		17.7	Air	Closed	Direct air injection.	Single unit fuel pump and injector.	Reduction gear 2:1 - Speed 1300 r/m - Brake H.P. 150 - Weight 1760 lbs.
V Deutz. Cologne. a	FMV 117	50	1250	4	4.5	6.7		26.9	Solid	Closed	Straight flow precombustion.	Single unit variable stroke plunger.	In large production - One of the 5 ton trucks recently run from Cologne to Berlin in motorboats.
	FMS 117	75	1250	6	4.5	6.7		21.8					
a Korting		90	1200	6	5.1	7.5		22	Solid	Closed	Precombustion chamber.	Single unit constant stroke plunger.	Experimental unit operating nicely to make delivery
VI Krupp. (Kiel) a	F 14	65	1000	4	5.1	7.2	16		Solid	Open	Direct injection.	Single unit constant stroke bypass valve.	Six just delivered German Admiralty for motor boats. Completed rapid tests in auto trucks.
	F 16	100	1000	6	5.1	7.2	15						
	F 26	200	1000	6			13.2						
VII Lieke-Hoffmann Buroh-Werke. a	A 4	60	1500	4	4.5	6.5		26.1	Solid	Closed	Direct injection.	Combination pump valve in variable stroke plunger. Lead control multiple unit.	In production in at least five sizes. These engines are the only four stroke engines employing "multiple unit" fuel pumps.
	A 6	90	1500	6	4.5	6.5		23					
a	B 4	100	1200	4	5.7	8.3							
a	B 6	150	1200	6	5.7	8.3							
VIII Beardmore (England)		100	1200	4	6	8	18		Solid				Two million miles in real use, larger plug at R.H. Type installed and favorably reported by Canada National Railways.
		150	1200	6	6	8	16						

a R.P.M. & Power may be reduced about 10% in motor boats where automotive values are given. For example see I. M.A.N.

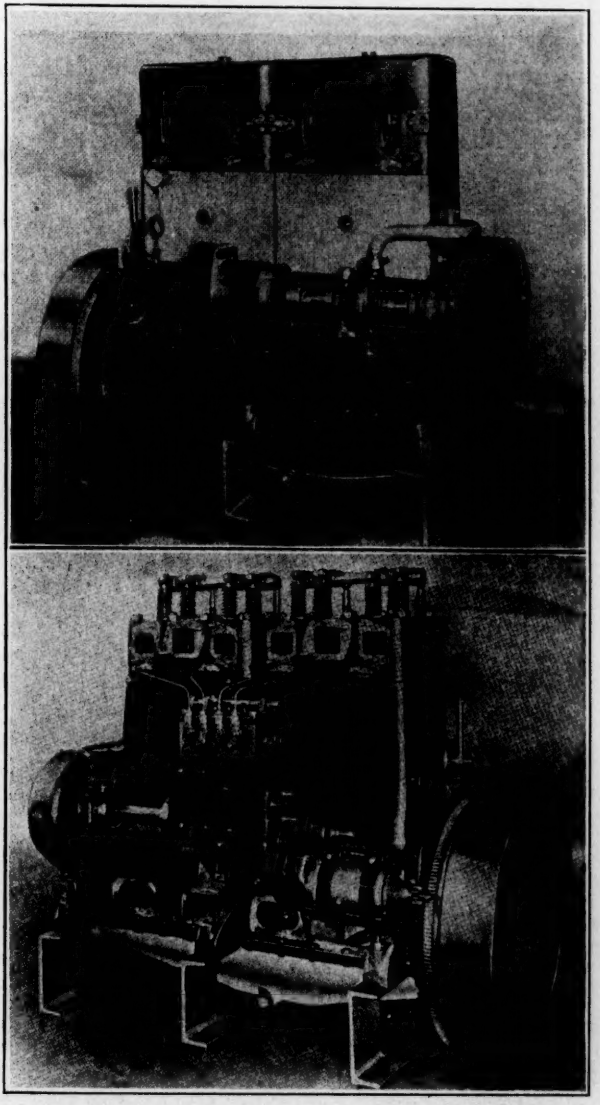


FIG. 26.—MASCHINENFABRIK AUGSBURG NUERNBERG M. A. N.

STANDARDIZATION OF PACKING MATERIALS FOR NAVAL AND MARINE MACHINERY.

By F. M. McGeary.*

The manufacture and installation of machinery for power plants handling steam, water, oil, gasoline, acids and other liquids or gases offers a diversity of uses for materials that will effectually seal joints against leakage. Of the variety of joints which might be considered there are three main types of primary importance, those which slide such as expansion joints, piston rods, pistons; those which rotate such as shafts and those which are fixed such as flanges and bonnets. Of these, the moving joints offer the greatest difficulties since it is necessary to seal against leakage without causing excessive friction and consequent power consumption. In addition to power consumption, friction also causes undue wear on the rod and early failure of the packing.

Some idea of the effect of friction upon power consumption may be obtained from tests made in an apparatus especially constructed for this purpose. This apparatus consists of a double ended stuffing box and a reciprocating rod. The reciprocating rod receives its motion from a cross head and a crank arm connected to a worm and gear on a 2.5 H.P. electric motor. The double ended test stuffing box is a cylindrical shaped casting ten inches long, with a steam space in the center and a stuffing box at each end. The steam space in the center is slightly larger in diameter than the stuffing boxes which are 3¼ inches deep. The reciprocating rod is 2 inches in diameter and is made of cold rolled steel. A piston valve alternately admits and releases steam of 275 pounds per square inch pressure to and from the stuffing boxes during each stroke of the reciprocating rod. The reciprocating rod operates at a speed of 75 strokes per minute with a stroke of 12 inches. The speed and power of operation of the

* Materials Engineer, Supply Division, Bureau of Engineering, Navy Department.

apparatus are taken by hourly readings of a stroke counter, a voltmeter and an ammeter. In order to determine the amount of power consumed by frictional resistance of the packing, a zero load, or the power required to operate the apparatus with no packing in the stuffing boxes is determined before and after the test. The test is run for a period of about 150 hours.

The results in Table I, which are the average of a number of tests, give the frictional resistance in foot pounds per minute for two standard steam packings. In this tabulation, a symbol number which will be described later in Table II and III is used to identify the type and kind of packings.

TABLE I.

Symbol	Average Friction Foot-pounds per minute	Average Friction H. P.	Approximate Life of the packing in hours
1400	10,000	0.304	8000
1100	22,000	0.660	2100

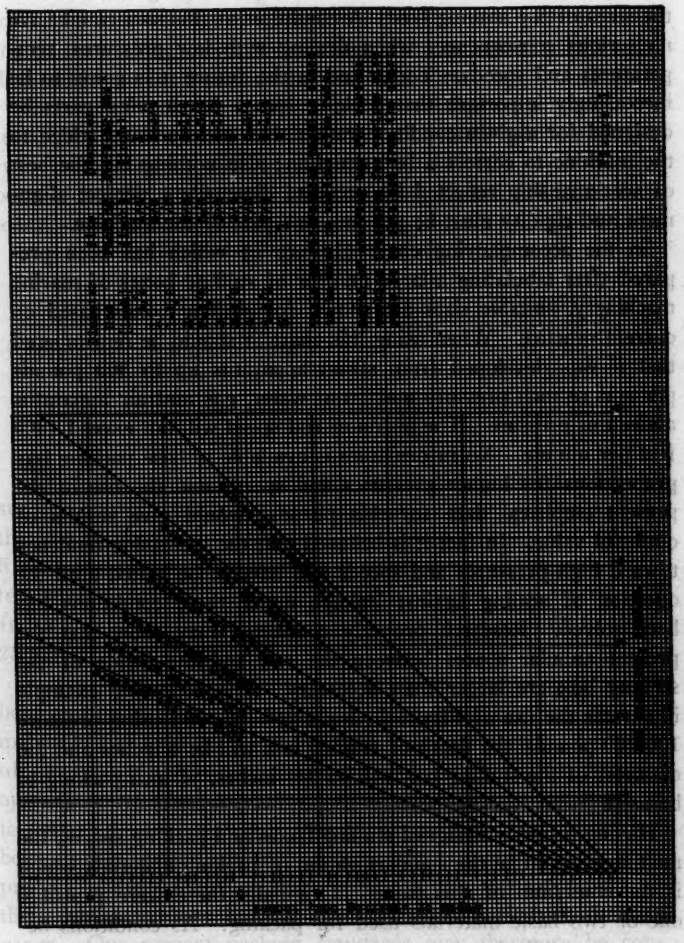
Under the conditions of this test where the maximum rubbing surface between the rod and packing is about 47 square inches, the loss in power is considerable. The effect of friction on the life of the packing is shown by a comparison of the endurance test results obtained on symbols 1400 and 1100. An increase in friction of about 120 per cent cuts the life of the packing to about 25 per cent.

In the early days of power plants when pressures were low both as to steam and the liquids or gases being handled, packing materials and packing spaces were of little importance. In many of the earlier installations packing spaces and methods have the appearance of being emergency measures added to the machine to correct a situation which was giving trouble. As leakage has been an ever present evil and its correction a solution for the then present moment many of these early packing measures with all their crudeness have become standard packing practice through usage. Our present packing standards may therefore be considered to be the result of trial and error in packing against low pressures and not so far as can be determined the result of any careful study of the question.

This seems to be a fair conclusion after examination of the plans for a number of units of different manufacture having the same pressure and rod sizes when one finds a large variation in the sizes of the stuffing box. A condition of this sort could have no serious consequences when dealing with low pressures, temperatures and speeds. Today and in the future it is going to be increasingly important that packing spaces for moving parts be designed for a given condition rather than merely being added to the machine to correct a bad situation; also nothing but the best of workmanship reflected by accurate dimensions, perfect alignment and rigidity must be put into this part of the machine.

In the accompanying chart (Figure I) which has been prepared after tabulating a large number of stuffing boxes of marine machinery, suitable sizes are shown for stuffing boxes based on current practice. When using this chart, it should be remembered that it was developed from steam apparatus up to 300 pounds per square inch pressure and from water, gas and air apparatus above this pressure. The piston or rod speed will not exceed 1800 feet per minute. No information is available to base packing box proportions for higher pressures and speeds and the use of the present empirical data for this purpose would give results of doubtful value. In fact it is believed that the increasing tendency toward high pressures and speeds will have to be halted until the question of sealing against leakage under these conditions has been solved. Research properly conducted may show that the present annular spaces between the rod and the inside of the stuffing box are larger than necessary. The following example illustrates the use of Figure I; for a 2½ inch rod using pressure 100-200 pounds, the depth of stuffing box will be 4 inches and the diameter of the stuffing box will be 3¾ inches, ⅝ inch packing to be used.

Having properly designed stuffing boxes and packing spaces, the next step would be the standardization of packing materials which is the chief aim of this paper. Originally there were probably one or two basic materials used for packing. As conditions became more difficult other materials were used and eventually the manufacture of packing on a commercial basis was started. Successful marketing of packing materials by one commercial con-



cern caused others to enter the field. Competition started and with it brand or trade names came into use. The sale of packing became more a matter of selling brands of packing than the sale of a specific kind of packing. While this may have been so it is not the intention to infer there is or was anything dishonest in this practice. It seems to have been a natural evolution in the business, and as long as a brand or trade name represented a given standard of quality no fault can be found with the practice. Unfortunately all sellers of packing are not manufacturers; and a brand of packing might and did vary from time to time even though it was the honest desire of the seller to duplicate previous results by a different manufacturing source.

Brand or trade names were then the first descriptions of packing materials and they are used to a large extent for this purpose today; in fact to a much larger extent than there is any need for with our present knowledge of the most used materials. Fundamentally we have the following, all of which have been successfully used as packing materials over a period of years; asbestos fiber, carbon, leather, metal, rubber, and vegetable fibers. A brief description of the various basic materials is given in order to provide a background for the standardization which will be proposed later.

Asbestos is the fibrous form of Hornblende. The asbestos commonly used in the packing industry is that known as "Chrysotile." It separates, on crushing, into white cotton like fibers. The province of Quebec in Canada is the principal source of supply, although quantities are produced in Russia, Rhodesia and Arizona. It is commonly supposed that asbestos is immune from the effects of heat, however high the temperature. While it will resist the action of heat to a remarkable degree, it will finally break down at elevated temperature. This variety of fiber has high tensile strength and considerable flexibility both of which are due to its high content of combined water. Ability to retain its combined water gives asbestos its value as a packing material. Asbestos fiber is used in packing in the form of twisted or braided yarns, after weaving into cloth, and in the form of felted fiber sheets. Of the fiber or fabric packings it is by far the most important.

Carbon as a packing material can be used only in the form of fixed rings. It is made of calcined petroleum coke, graphitized

scrap and pitch, the materials being mixed and extruded through a die and baked at a high temperature. It is then broken up and ground to pass a fine mesh screen and again mixed and ground to pass a somewhat coarser mesh screen after which it is molded cold under high pressure into ring or rectangular shape. The molded shapes are baked at a medium high temperature, after which they are machined to the size of ring desired. A ring is generally divided into two or more segments to permit compensation for wear. Graphite is another form of carbon used more or less in packing, but its chief purpose is that of a lubricant.

Leather is a substance prepared from the skin of animals by chemical and mechanical processes. For packing purposes the leather most used is that produced from the hides of steers and cows, tanned with oak bark, or a combination of vegetable tanning materials. Only full grained leather with the flesh side smooth and free of loose flesh and which is shaved or otherwise leveled to an even thickness is commonly used for packing purposes. Leather is used chiefly for hydraulic purposes in the form of cups and as a packing material is of little importance.

Metal is used as a packing material in a number of ways. It may be a casting, machined to ring form for use in floating ring packing of reciprocating machinery for which purpose both cast iron and red bronze is used, or as a spring ring on the pistons of pumps and internal combustion engines for which purpose cast iron is used exclusively. It is also used in sheet form as a gasket material for which purpose iron, copper, lead, and aluminum are used. It may be used in foil or shredded strand form, the foil or strands being fashioned into square rope or molded into rings for which purpose lead and aluminum are used. It may be used in the form of small particles held together in a common mass by binders of mineral or vegetable origin for which purpose lead aluminum and copper are used. Metallic packings will probably be the chief material used for packing the higher pressures and temperatures on account of its low frictional resistance when properly installed.

Rubber is a vegetable product obtained by a process of coagulation from a milky fluid called "latex" contained in the cell system of a number of trees, vines and shrubs. The rubber, which is suspended in the serum, is separated by evaporating part of the

water; when it is purified and dried and is ready for the market. There are two sources of rubber, wild and plantation. Central and South America yield the bulk of wild rubber and establish the quality factor. "Para" a variety known as "Hevia" is named from a province in Brazil and is the best grade and the most suitable rubber for packing materials. Two-thirds of the world's supply is of this variety and it is now cultivated, being grown on plantations in Ceylon, Malay States, Dutch East India, Borneo and some Pacific Islands. Plantation rubber has all the characteristics of wild para and the additional advantage that under cultivation, the production is systematic and the tree is preserved. Crude rubber is chiefly a hydrocarbon compound. It is soft, solid and elastic. It also contains varying percentages of resins and proteids, neither of which are desirable. On warming, unvulcanized rubber becomes soft and sticky and melts. Its technical value depends upon its elasticity, *i.e.*, ability to stretch without breaking and to return to the original form upon removal of the stress.

There are many formulas used in the compounding of rubber and it is remarkable the number of combinations that will produce similar results. While it is customary to speak of the various ingredients added to rubber in compounding as "fillers" this is not strictly correct because rubber alone is of very little commercial value. It becomes a commercial asset only after certain ingredients are added. These are of two classes, those that impart desirable qualities and those that are added merely to increase the weight or cheapen the mixture. Compounding ingredients are known as fillers, accelerators and pigments, the most commonly used of which are zinc oxide, lithophone, litharge, lampblack, etc. These ingredients are mixed with the rubber in proportions for the intended purposes. Sulphur is also added to assist in vulcanizing or curing to give stability to the finished product.

Rubber is not a suitable packing material when used alone except for water conditions under low pressures. However, practically all kinds of packing materials use rubber in some form; hence it may be said to be one of the most important of the packing materials.

Vegetable fibers in the order of their importance for packing purposes are cotton, flax, jute and hemp. Cotton is the fiber

attached to the seed of the cotton plants, cultivated extensively throughout the world. Physically the individual cotton fiber consists of a single long tubular cell having the appearance under the microscope of a broad, finely grained band, which is repeatedly twisted about its axis. This natural spiral like twist causes it to be especially adaptable to purposes of spinning. The spinning quality of cotton fiber depends on both the nature and amount of twist which causes the individual fibers to lock themselves together, and the length of fiber. For packing, cotton fiber is mixed with asbestos fiber to act as a binder and facilitate spinning of the asbestos. It is also used in the form of sheeting and duck as a reinforcement for rubber packing. It is hardly ever used alone as a packing material on account of its tendency to disintegrate in contact with moisture.

Flax is the term used to denote both a plant and the fiber from it. Flax produces the fiber from which the linen of commerce is manufactured and is a cultivated annual plant of erect stalk rising to a height of from 20 to 40 inches. It is grown very generally in a number of varieties all over the Northern Hemisphere, the best grades coming from Russia and Belgium. To produce the fiber, the stalks are pulled up by the roots and immersed in water until the membrane or rind becomes loose. They are then removed from the water and dried after which the fibers are separated by flailing or pounding with a wooden mallet or similar device. The fibers near the rind are coarse and short and are known as tow while those near the center are fine and silky in appearance and are the long line fiber used in the manufacture of packing. Flax is graded by length of fiber and is known as "long line," "line" and "tow," the length averaging 22, 17 and 14 inches, respectively. There is also a grade known as "cut" which includes all broken and short lengths arising from handling the major grades. This grade is frequently improperly called "tow." In the fiber, flax and jute are very similar in appearance to the naked eye and can only be distinguished by the aid of the microscope. For this purpose, the fibers are stained with a chloriodide solution and examined. Flax shows cross markings on the fiber while a distinct yellowish coloration and no cross markings indicate jute. Jute is a harsh and brittle fiber breaking much more readily than

flax. Under damp conditions, it decomposes more rapidly than flax. Jute and hemp except for length of staple are similar in characteristics to flax. Flax and jute are used as packing materials in the form of square braided rope made directly from the fibers. Jute and hemp are used to make paper stock later used as packing after special water proofing processes.

While one or more of these basic materials are to be found in packing for every purpose, it is not considered possible to reduce the number of packings to the materials described. The ideal situation would be to have only one type of rod and one type of sheet packing for all purposes. This is impractical due to the fact that none of the basic materials are suitable for all purposes and different types have been developed to meet the special requirements of the various general service conditions. This can and has been carried to the other extreme of having a special type not only for a given service condition but for each kind of equipment. The situation is further complicated by the tendency to specify packing for a particular purpose by its proprietary brand name. From an examination of the catalogs of three of the largest distributors of packing 130, 105 and 75 types, respectively, are found to be described. As there are in each case a large number of special types peculiar to only one of these concerns, it is safe to assume there are more than 1000 types of packing available for purchase by brand name for the general conditions in a marine power plant. Closer examination of the various types offered under brand names by the packing industry will show certain types common to sellers. It will also show that a large number of the so-called special types are simply a combination of two or more of the common types and that the same effect can be produced by the user if he will drop the higher priced combinations making the same combinations by using the less expensive common types. Standardization and simplification will reduce costs to both the seller and the user without reducing the quality of service to be obtained.

In the interest of simplicity, to avoid possible errors in nomenclature and to obviate the necessity of using proprietary brands as a description of the kind of packing desired, a symbol system has been devised to identify each kind and type of packing considered necessary for use in any power plant. This symbol is a four digit

number in which the first digit indicates the class of service, *i.e.*, moving, such as rods, shafts, valve stems, etc., and fixed, such as flanges or bonnets. The second digit indicates the predominant material of which the packing is composed, *i.e.*, asbestos, vegetable fiber, rubber, metal, etc. The third and fourth digits indicate the different style or forms of packing made of the predominant material. Table II shows the symbol make up in the first six columns and the completed symbol in the seventh column.

For the purpose of standardization, this symbol number is in effect a specification taking the place of brand or trade names and also eliminating the use of the present haphazard nomenclature. A symbol system such as this when firmly established should be all the information required by a power plant operator when ordering packing from any supplier for replacement providing he prepares a packing chart similar to Table IV from the information available in Table III. In both of these tables a number of symbols are shown as satisfactory packing for a given service. When more than one symbol is shown for a given service, each has been found by service test to be suitable for the required service. This range of choice provides greater flexibility in caring for packing needs either in case of special packing problems or of depleted storerooms.

Table III is the "Table of Recommendations" for use in each of the various services. Table IV is a suggested chart for a marine power plant which is applicable in form for any power plant. Such a chart should be prepared for the ready use of the operating personnel.

This plan of symbolization was devised in the Bureau of Engineering to standardize and to simplify the procurement of packings for the machinery of Naval Vessels. It will be used hereafter in conjunction with the "Table of Recommendations" for Supply and Operation.

A brief description of the various symbols used in these tables follows in numerical sequence. In this description the usual designation of the packing is used but the tendency should be toward the abolition of this and to the use of symbols entirely.

Symbol 1100 is an "asbestos high pressure rod" packing made from woven cloth treated with a rubber compound and wrapped

TABLE II.

DIGIT	CLASS OF SERVICE	DIGIT	KIND OF MATERIAL	DIGIT	STYLE OF PACKING	COMPLETED SYMBOL
1	Moving	1	Asbestos	00	Asbestos High Pressure Rod	1100
				01	Asbestos Valve Stem	1101
				02	Asbestos Wick and Rope	1102
				03	Braided Asbestos Rod	1103
2	Fixed	1		50	Compressed Asbestos Sheet	2150
				51	Asbestos Metallic Cloth Sheet	2151
				52	Asbestos Metallic Cloth Gaskets	2152
				53	Asbestos Metallic Cloth Tape	2153
				54	Asbestos Cloth Sheet	2154
1	Moving	2	Vegetable Fiber	00	Cotton Wick	1200
				01	Cotton Condenser Tube	1201
2	Fixed			26	Hard Fiber Sheet	2226
				27	Chart Paper, Shellaced	2227
1	Moving			60	Braided Flax	1260

1	Moving	2	Vegetable Fiber	00	Cotton Wick	1200	
				01	Cotton Condenser Tube	1201	
2	Fixed			26	Hard Fiber Sheet	2226	
				27	Chart Paper,Shellaced	2227	
1	Moving			60	Braided Flax	1260	
2	Fixed			90	Plant Fiber Sheet	2290	
1	Moving	3	Rubber	00	Low Pressure Rod	1300	
				01	Tucks Square	1301	
				02	Tucks Round	1302	
				03	Tucks Rock Hard	1303	
				04	Rubber Strip	1304	
2	Fixed			51	Rubber Sheet,Vulcanized	2351	
				52	Rubber-Graphite Sheet,Unvulcanized	2352	
				53	Rubber Sheet,Cloth Insertion	2353	
				54	Rubber Sheet,Wire Insertion	2354	
1	Moving	4	Metal and Fabric	00	Semi-metallic Rod	1400	
2	Fixed			10	Copper Asbestos Gaskets	2410	
				11	Monel Metal Asbestos Gaskets	2411	
				12	Steel Asbestos Gaskets	2412	
				13	Copper cork gaskets	2413	
1	Moving	4	Metallic	30	Flexible Metallic Rod	1430	
				31	Plastic Metallic Bulk	1431	
				32	Plastic Metallic Metal Case	1432	
				33	Plastic Metallic Fabric Case	1433	
				34	Shredded Lead	1434	
				35	Flexible Metallic Condenser Tube	1435	
				36	Braided Copper Wire	1436	
				40	Cast Iron Rings	1440	
				41	Spring Rings,Cast Iron	1441	
				42	Bronze Rings	1442	
				43	Babbit Rings	1443	
1	Moving	4		50	Cage Type,Cast Iron Rings	1450	
				51	Cage Type, Bronze Rings	1451	
				52	Cage Type, Babbit Rings	1452	
2	Fixed			70	Soft Iron Sheet	2470	
				71	Aluminum Sheet	2471	
				72	Copper Sheet	2472	
				73	Lead Sheet	2473	
2	Fixed	5	Joint Compound	00	Furmanite	2500	
1	Moving	6	Leather	00	Cup Leathers	1600	
1	Moving	7	Carbon	00	Carbon Rings	1700	

TABLE III.

PACKINGS

TABLE OF RECOMMENDATIONS.

Service	For Sliding or moving joints								For Flanges and bonnets fixed joints		
	Pounds per square inch Gauge Pressure	Solid Compressible Material, Fibre, Asbestos rope, Fibrous, and Cotton, metal plungers	Metal rods and slides	Fibre glands or rods (Lubricated metal)	Segmental or slip joints	Plungers and plungers (Lubricated metal)	Cast-iron tube	Permanent Lubrication	Temporary Lubrication	Working castings
			1409	1482	1101	1109	1440	1486	2160	2160	2262
			1100	1482	1480	1480	1441	1201	2161	2161	8000
Steam, saturated, low pressure	Up to 150	875	1481	1108	1481	1486			2188	2188	
Steam, superheat, low pressure	Up to 150	880	1480	1700	1400	1488			9470		
Steam, saturated, high pressure	150 to 800	800	1109	1100	1108	1440			9410		
			1481								

Steam, saturated, low pressure	Up to 150	975	1400 1100 1481	1481 1408	1101 1480	1100 1480	1440 1441	1438 1801	2180 2181	2180 2181	2862 2800		
Steam, superheat, low pressure	Up to 150	880	1481	1108	1481	1436			2188	2188			
Steam, saturated, high pressure	150 to 300	800	1450 1109 1481	1700 1100	1400 1108	1438 1440			2470 2410				
Steam, superheat, high pressure	150 to 450	725	1400 1481 1432 1481 1100 1100	1481 1408 1108 1700 1100	1101 1480 1481 1108 1108	1432 1436 1481 1480 1442	1440 1441	1438 1801	2180 2181 2188 2188 2153 2470	2180 2181 2188 2188	2800 2862		
Water, cold, fresh and salt, low and high pressure	Up to 250	180	1260 (AA) 1400	1260 1481 1400	1260 1101 1481		1448 1208 1801	1438 1801	2264 2181 2180	2263			
Water, bilge, cold	Up to 50	88	1208 1501 1100	1488	1480 1488 1108				2265 2478 (DD)				
Water, hot, fresh	Up to 200	880	(AA) 1260 1100	1481 1480 1488	1260 1101 1480		1448 1208 1801	1438 1801	2181 2266 2264	2264			
Water, hot, salt	Up to 150	880	1200	(AA)	2488		(CC)		2478				
Water, hot, brine	Up to 100	880	1488 1480	1260 1108	1108				(DD)				
Air, low pressure	Up to 800		(BB) 1500	(BB) 1481	1101 1480		1440 1441		2180 2264	2264 2180			
Gas, Sulphur dioxide	Up to 75		1100	1488	1488				2182				
Gas, Ethyl chloride	Up to 80		1482	1108	1488				2478				
Gas, Ammonia	Up to 75		1481 1400	1101 1100	1400 1108				2266				
Air, high pressure	Up to 3000		(BB) 1400 1480	(BB) 1480 1400	1101 1480 1481		1440 1441		(EE) 2181 2180	2181 2180 2188			
Gas, Carbon dioxide	Up to 3000		1100 1481 1480	1488 1108 1100	1488 1400 1108				2188 2266	(EE)			
Gases of combustion	Up to 500	725			1484 1480 1481		1441		2290 2180 2227 2471 2410	2290 2181 2180			
Oil, fuel, low pressure	Up to 300	180	1400 1481 1480	1481 1480 1488	1260 1101 1480		1448 1440		2290 2180 2227	2290 2180 2267			
Oil, lubricating	Up to 80	800	1482	1260	1481				2471				
Gasoline	Up to 80	180	1481 1100	1400 1108	1400				2478				
Oil, fuel, high pressure	Up to 800	250	1400 1481 1480 1482 1481 1100	1481 1480 1482 1260 1400 1108	1101 1480 1400 1481 2488 1108		1448 1440		2471 2188 (EE) 2470 2180 2478	2180 2471			

(AA) Use alternate rings of symbols 1260 and 1208
(BB) Use alternate rings of symbols 1400 and 1480
(CC) For bottom blow valves discharge. Use rings cut from symbol 2180 of 1/16" thickness
(DD) Use symbol 2801 for water tight packing without pressure
(EE) Ground Joints

Table IV.

Machinery Index No.	UNIT	LOCATION OF PACKING	USE SYMBOL NO.
241-1	Main turbines	Casing joints, high pressure	2500-2558
		Casing joints, low pressure	2558-2800
		Shafts, stuffing box	1700-
		Control Valves, stuffing box	1101-1430-1431-1105
241	Turbo-generator	Turbine shaft, stuffing box	1431-1432-1105-1700-1100
		Turbine casing joints	2558-2800
251	Boilers	Uptake doors and casing joints	2154-2151-2150
		Gauge glasses	1101-1430-1431-1105
		Steam flange fittings	2150-2151-2158-2470-2410
		Water flange fittings	2151-2254-2154-2472
		Manholes and manhole plates	2152-2411-2410-2412
		Burner holders	2472
		Bottom blow valves, plungers	2150
263	Main blowers, Turbine	Turbine shafts, stuffing box	1431-1432-1105-1700-1100
		Control Valves, stuffing box	1101-1430-1431-1105
258-3	Heaters, Fuel Oil	Manifolds	2250-2150-2247-2471-2472
264-1	Heaters, Feed water	Heater joints	2151-2254-2154-2472
247	Main air pump, reciprocating Vertical triplex, 18 x 30 x 24"	Steam rod, stuffing box	1400-1100-1431-1430-1105-1431
		Service rod, stuffing box	(BB)-1300-1100-1432-1431-1600
		Valve stem, stuffing box	1101-1430-1431-1400-1105
		Steam piston	1440-1441
		Plunger, wet or dry end	1442-1302-1301
247	Main and auxiliary feed pump, reciprocating Vertical simplex, 16½ x 11 x 18"	Steam rod, stuffing box	1400-1100-1431-1430-1105-1431
247	Distillers fresh water pump, reciprocating Vertical simplex, 3½" x 4 x 4"	Service rod, stuffing box	(AA)-1300-1100-1300-1432-1430
		Valve stem, stuffing box	1101-1430-1431-1400-1105
		Steam piston	1440-1441
247	Evaporator feed pump, reciprocating Vertical simplex 4½ x 6 x 6"	Water plunger	1442-1302-1301
247	Fire and bilge pump, reciprocating Vertical simplex 7 x 7 x 12"	Steam rod, stuffing box	1400-1100-1431-1430-1105-1431
		Service rod, stuffing box	1300-(AA)-1400-1302-1301-1100
		Valve stem, stuffing box	1101-1430-1431-1400-1105
247	Lubricating oil cooler pump, reciprocating Vertical simplex 8 x 9 x 12"	Steam piston	1440-1441
		Water plunger	1442-1302-1301
247	Lubricating oil pump, reciprocating Vertical simplex, 7 x 8 x 12"	Steam rod, stuffing box	1400-1100-1431-1430-1105-1431
		Service rod, stuffing box	1400-1431-1430-1432-1431-1100
		Valve stem, stuffing box	1101-1430-1431-1400-1105
247	Fuel oil booster pump, reciprocating Vertical simplex 6½" x 7 x 8"	Steam piston	1440-1441
		Oil plunger	1442-1440
247	Fuel oil service pump, reciprocating Vertical duplex 6½ x 6 x 8"	Steam rod, stuffing box	1400-1100-1431-1430-1105-1431
		Service rod, stuffing box	1400-1431-1430-1432-1431-1100
		Valve stem, stuffing box	1101-1430-1431-1400-1105
		Steam piston	1440-1441
		Oil plunger	1442-1440

	Vertical simplex 6" x 7 x 8"	Oil plunger	1448-1440
847	Fuel oil service pump, reciprocating Vertical duplex 6½ x 4 x 6"	Steam rod, stuffing box Service rod, stuffing box Valve stem, stuffing box Steam piston Oil plunger	1400-1100-1451-1450-1105-1451 1400-1451-1450-1455-1451-1100 1101-1450-1451-1400-1105 1440-1441 1448-1440
847	Auxiliary air pump, reciprocating Vertical, 6½ x 10 x 8"	Steam rod, stuffing box Service rod, stuffing box Valve stem, stuffing box Steam piston	1400-1100-1451-1450-1105-1451 (2B)-1500-1100-1455-1451-1400 1101-1450-1451-1400-1105 1440-1441
847	Auxiliary circulating pump, reciprocating Crank case cpring, 5 x 8	Steam rod, stuffing box Valve stem stuffing box Steam piston Rotary shaft stuffing box	1400-1100-1451-1450-1105-1451 1101-1450-1451-1400-1105 1440-1441 1500-1451-1450-1455
847	Main circulating pump, rotary Centrifugal, turbine drive	Turbine shaft, stuffing box Service shaft, stuffing box	1451-1452-1105-1700-1100 1500-1451- 1450-1455
849-1	Air compressor, reciprocating high pressure, four stages	Steam rod, stuffing box Valve stem, stuffing box Service rods, stuffing box Plungers, all stages Steam piston	1400-1451-1451-1450-1105 1101-1450-1451-1400-1105 (2B)-1600-1650-1100-1451-1450 1440-1441 1440-1448
847	All pumps, Miscellaneous	Steam end, cylinder heads Oil end, cylinder heads Water end, cylinder heads Valve cages	2150-2151-2158-2470-2410 2290-2150-2227-2471-2472 2264-2151-2150-2265-2473 2473
858-1	Evaporator	Headers	2150-2151-2158-2470-2153-2410
858-2	Distiller	Headers	2150-2151-2158-2470-2153-2410
848-1	Condenser	Headers Tubes	2150-2151-2158-2470-2153-2410 1435-1801
848	Piping, steam	Valves, stuffing box Flanged joints Expansion joints	1101-1450-1451-1400-1105 2150-2151-2158-2470-2410 1100-1450-1456-1458-1440
848	Piping, saltwater, cold Fresh water, cold Bring, cold	Valves, stuffing box Flanged joints	1250-1101-1451-1450-1455-1105 2264-2151-2150-2265-2473
848	Piping, Salt water, hot Fresh water, hot Bring, hot	Valves, stuffing box Flanged joints	1250-1101-1450-1455-1105 2151-2264-2154-2473
848	Piping, Air	Valves, stuffing box Flanged joints	1101-1450-1451-1455-1400-1105 2150-2264-2158-2475-2250
848	Piping, fuel oil, L.P. Piping, oil	Valves, stuffing box Flanged joints	1250-1101-1450-1451-1400 2290-2150-2227-2471-2472
848	Piping, fuel Oil, H.P.	Valves, stuffing box Flanged joints	1101-1450-1400-1451-1455-1105 2471-2158-(2)1-2470-2150-2472
852	Motor boat engines	Cylinder head Crank case Piston rings Spark plugs Intake manifolds Exhaust manifolds	2150-2151-2410 2290-2227 1441 2410-2472 2226 2151-2150-2410

tightly upon itself, in layers. It is made in three styles; of plain cloth without either rubber core or rubber back; with a rubber core; or with a rubber spring back. No rubber compound is used on the outside of the packing.

The asbestos cloth is made of yarn containing 90 per cent of asbestos of not less than 12 per cent water of composition. There are 20 strands in the warp and 10 strands in the filler. Each strand contains 2 asbestos yarns.

Only rubber of the best quality is used in the core, spring back and friction compound. There is not less than 60 per cent, by volume, of rubber in the spring back or central core. The total sulphur in the rubber compound, exclusive of that contained in the barytes is less than 8 per cent. When the packing contains a central core or spring back there is from 35 to 60 per cent of rubber in the packing, prior to lubrication. When no central core or spring back is used, there is from 25 to 50 per cent of rubber in the packing, prior to lubrication. The cross sectional area of the rubber core or the rubber spring back will not exceed 12 per cent of the nominal cross sectional area of the packing.

The lubrication will not exceed 20 per cent, by weight, of the finished packing. Either oil or glycerine is used as a lubricant, the latter being more generally employed.

When subjected to the action of saturated steam at 250 pounds per square inch gauge, for 8 hours, the finished packing will not harden nor soften materially, and the friction compound will retain most of its elasticity and strength as indicated by the "Tooth" when the plies are separated.

The finished packing is usually dipped in graphite after it has been coiled and set by slight vulcanizing. The inside diameter of the coil is from four to six times the thickness of the packing.

Symbol 1101 is an "asbestos" valve stem packing made from braided asbestos yarns which have been treated or sized with a heat resisting rubber compound.

The asbestos yarn contains not less than 90 per cent of asbestos of not less than 12 per cent water of composition. A single ply of yarn will weigh not more than 1 pound per 1000 yards.

The packing is not impregnated with lubricating materials; although it is dipped in flake graphite as a final treatment.

Symbol 1102 is an "asbestos wick or rope packing." It is made of long fiber carded asbestos, spun into rovings, and laid up into the general form of a common lamp wick or loosely twisted rope which can be easily separated. When a core is used it is of the same material and construction.

The asbestos rovings contain not less than 85 per cent, by weight of asbestos, of at least 12 per cent water of composition.

No wire insertion, sizing nor lubrication is used.

The weight per linear yard is not less than that shown in the following table:

Nominal Size	Weight per Linear Yard (Minimum)
Inches	Pounds
1/4	0.045
3/8	.12
1/2	.20
5/8	.30
3/4	.40
7/8	.50
1	.75
1–1/4	1.00

Symbol 1103 is a "braided asbestos rod" packing. It is made from long fiber asbestos rovings braided square and lubricated with high grade lubricating compounds. The rovings will contain 85 per cent of asbestos of not less than 12 per cent water of composition. The lubrication will not exceed 20 per cent by weight of the finished packing.

Symbol 1200 is a "cotton wick" packing. It is made of new cotton, thoroughly carded, and wholly free from seeds, lumps or foreign particles of any kind. The cotton is not bleached nor subjected to any chemical processes that change its natural condition.

The cotton is soft spun into rovings with 7 plies to the strand. The tension used in spinning is such as to produce a soft wick which will easily absorb a heavy cylinder oil.

Symbol 1201 is a "cotton condenser tube" packing. It is made of 20 strands of twisted cotton yarn, each yarn composed of 2 twisted threads; the 20 strands are closely braided around a cotton

core composed of 4 strands of yarn, each yarn consisting of 11 threads. The finished packing is about 1/8 inch in diameter and weighs about 1 pound per 200 yards.

Symbol 1260 is a braided flax rod packing. It is made from well cleaned long line fiber. There is no warp, core, or foreign fiber, or substitutes. After being braided and squared, it is impregnated with pure tallow.

The length of staple is such that when a section of packing about 20 inches long is carefully unravelled and combed out with a coarse comb, it will give a brush in which 40 per cent of the fiber, by weight, is 8 inches long.

The quality of the staple is such that when a roving is carefully untwisted and the fibers straightened out, the tensile strength in kilograms, will be 100 times the weight of the roving expressed in grams per centimeter of length.

The finished packing will contain from 35 to 50 per cent of tallow. The tallow is pure and refined, free from rancidity and will contain not more than the equivalent of 2 per cent of oleic acid.

The weight of the finished packing per linear yard will be as shown in the following tabulation:

Size of Packing	Weight per Linear Yard	
	Minimum	Maximum
Inches	Pounds	Pounds
1/8	0.0185	0.024
1/4	.083	.108
3/8	.185	.24
1/2	.324	.42
5/8	.508	.66
3/4	.74	.96
7/8	1.015	1.32
1	1.29	1.68
1-1/8	1.66	2.16
1-1/4	2.08	2.70

Symbol 1300 is a "low pressure spiral gland" packing. It is cut from slab stock built up of layers of rubber frictioned duck laid on the bias at 45 degrees angle and in alternate direction, each ply over-lapping the butted joint of the adjacent ply. Each slab has a top and bottom cover of rubber compound.

The duck fabric used in the plies weighs approximately 27.4 ounces per square yard. There will be one layer of frictioned fabric for each 1/16 inch of thickness, exclusive of the top and bottom covers. The top and bottom covers are rubber compound each about 3/64 inch in thickness. The rubber compound will contain not less than 45 per cent, by weight, of the best quality rubber, the acetone extract, corrected for free sulphur, of which will not exceed 6 per cent. The total sulphur, exclusive of that contained in the barytes will not exceed 8 per cent, by weight, of the rubber as compounded.

The slabs are cut into strips of square cross section, coiled into spirals and vulcanized, after which they are lubricated with a good grade of mineral cylinder oil and dipped in graphite. The quantity of lubrication will not exceed 30 per cent, by weight of the finished packing.

The packing will not show ply separation nor material hardening of the rubber compound when subjected to 4 hours exposure to steam at 50 pounds per square inch pressure or to a dry heat of 300 degrees F.

Symbol 1301 is a "square tucks" rod packing made of cotton duck frictioned with rubber compound. The packing is made as follows: 1/4 to 3/8 inch with 38 to 42 plies of fine duck, 1/2 to 7/8 inch with 26 to 30 plies of medium duck, and 1 inch and larger with 16 to 20 plies of coarse duck. The cotton duck layers are built up on the bias and the rubber compound is such that the packing will withstand boiling in water under 80 pounds per square inch steam pressure for four hours without ply separation or material hardening of the rubber.

Symbol 1302 is a "round tucks" rod packing made of cotton duck frictioned with rubber compound. It is similar to symbol 1301 except that it is round in cross section and contains a rubber core. The rubber core will contain not less than 75 per cent, by volume of rubber, the acetone extract corrected for free sulphur

of which is less than 6 per cent and the total sulphur exclusive of that contained in barytes is less than 8 per cent of the rubber as compounded. The rubber core will have a tensile strength of 900 pounds per square inch and an ultimate elongation of 300 per cent.

Symbol 1303 is a "rock hard tucks" rod packing similar to symbol 1301 except that it is vulcanized rock hard.

Symbol 1304 is a "rubber strip" packing made in sheets of the desired thickness and cut to the required width or it may be molded to the desired cross section. It is made entirely of rubber compound containing 80 per cent by volume of rubber.

The rubber compound will have a tensile strength of 2200 pounds per square inch, an ultimate elongation of 2 to 12 inches and a set of 20 per cent after a set stretch for 10 minutes of 2 to 11 inches. The tensile strength will not decrease more than 40 per cent after subjecting the packing to a dry air temperature of 160 degrees F. for 96 hours.

The weight of the packing will be 2.00 to 2.45 pounds per linear yard 1 inch square, other sizes being in proportion.

Symbol 1400 is a "semi-metallic rod" packing made of rubber frictioned asbestos cloth to which has been securely fastened a metallic member that becomes a continuous face when applied to the rod.

The asbestos cloth contains 18 strands in the warp and 9 strands of asbestos yarn in the filler. The asbestos yarn contains 85 per cent of asbestos fiber of 12 per cent water of composition.

The metallic member which is usually made of lead bar or foil amounts to about 60 per cent by weight of the packing.

Symbol 1430 is a "flexible metallic" rod packing made of wrapped, twisted, or braided metal foil, or of twisted shredded metal. It is usually made of lead although to some extent aluminum and copper is employed. It may be built around a core composed of hemp or asbestos yarn or it may have these materials embedded in the rovings or strands.

In this packing the entire wearing surface is of a metal which will not melt or oxidize below 550 degrees F. It is usually customary to add lubricating material, but this will not exceed 5 per cent of the weight of the packing.

Symbol 1431 is a "plastic material" in bulk form and suitable for use with any size of rod or in any annular space. It is prepared by mixing fine particles of metal, asbestos fiber and heavy petroleum wax. In some cases the wax is omitted and rubber is used. Metal may also be omitted and graphite used instead. Lead is the metal usually used although aluminum or copper is also employed. When metal is used the customary method is to use cut particles of strands about 3/4 by 1/8 inch of foil thickness. Lead shot may also be used. When metal is used it will amount to about 75 per cent, by weight, of the packing. When no metal is used about 50 per cent, by weight, of the packing is asbestos fiber.

Symbol 1432 is a "plastic metallic metal case" packing material similar to symbol 1431 which has been extruded into metal jackets of various sizes to produce a square packing. The metal jacket is generally of copper mesh construction, although it may be only wrapped in lead foil. The principal advantage of this type is in its ease of application and where the copper mesh jacket is used the packing material oozes through to the surface and the metal jacket acts as a reinforcing agency.

Symbol 1433 is a "plastic metallic fabric case" packing material also similar to symbol 1431 to which has been added a loosely braided cotton jacket or which has been extruded into a cotton knit jacket. In either case the jacket only serves to make use of the packing easier.

Symbol 1434 is a "shredded lead" packing made of strands of shredded lead twisted into the form of a rope and impregnated with a mixture of flake graphite and oil. The finished packing is 3/8 inch in diameter and 10 inches long. The lead strands are 3/32 inch wide and 1/64 inch thick. The lead alloy contains 5 per cent tin and 1/4 per cent antimony.

Symbol 1435 is a "flexible metallic condenser tube" packing. It is made of lead foil about 1 inch wide and approximately 4/1000 inch thick. The foil is wound on a mandril and on itself to the desired thickness, placed in a die and reduced by pressure to the required dimensions. It may also be made by twisting foil of the dimensions stated above into a roving winding the roving into the form of a helix on a mandril, placing in a die, and reducing by pressure to the required dimensions. The ring formed by either

process is endless. The foil which is practically pure lead has a small percentage of tin or antimony to increase its workability. Two metal rings and one fiber ring make a set of packing. The fiber ring is of the same material as described below under symbol 2226.

Symbol 1436 is a "braided copper wire rod" packing. It is made of rovings composed of a large number of very fine copper wires, the rovings being square braided after the fashion used to produce symbol 1260. No lubrication or material other than copper wire is used in this packing.

Symbol 1440 is a "cast iron fixed ring" packing which is used for various purposes, as follows: as the packing ring in the float-type or split case type of mechanical packing for which purpose the ring is divided into two or more segments held in place by a circumferential spring or other means, as a plunger ring where a split ring is fitted to a plunger and held in place by a follower plate or by a bull ring.

The rings will be made from a good grade of soft grey cast iron. The graphite carbon will be 2.75 to 3.00 per cent, the brinell hardness 75 to 100 and the tensile strength 3000 pounds per square inch.

Symbol 1441 is a "cast iron spring ring" commonly known as a "piston ring" for use on the pistons of internal combustion engines used on motor boats, motor cars, aeroplanes and diesels, and of air compressors, for which purpose it is used as a spring ring held in place by its own tension.

For these purposes, a good quality of grey iron is used. The graphitic carbon will be from 2.5 to 2.75 per cent, the brinell hardness will be 180 to 200 and the tensile strength will be about 3000 pounds per square inch.

Symbol 1442 is a "bronze fixed ring" packing used in the same manner as described for symbol 1440. It will be made of a non-ferrous composition containing a high percentage of lead. The following compositions are typical:

	No. 1	No. 2
Copper	75 per cent	39 per cent
Lead	20 per cent	60 per cent
Tin	5 per cent	0 per cent
Sulphur	0 per cent	1 per cent

Symbol 1443 is a "babbit metal fixed ring" packing used on the same manner as described for symbol 1440. It will be made of a high lead babbit, the following composition of which is typical:

Copper	4 per cent
Antimony	8 per cent
Lead	88 per cent

Symbol 1450 is a "cage type" or "split case type" of floating ring packing. It requires a special packing space. The cage or split case is made to fit in the packing space or it may be bolted on outside thereof. The packing cage is provided with three or more grooves, each of which will accommodate two split or segmental cut rings. The rings are held together and to the rod with a spring tension generally obtained by a circumferential coiled spring. The inside diameter of the packing groove being somewhat larger than the outside diameter of the packing ring when assembled permits a slight lateral motion in the packing which gives a so-called floating effect. The packing rings used with this packing are those described under symbol 1440.

Symbol 1451 is a "cage type" or "split case type" of packing similar to that described for symbol 1450 except that rings described under symbol 1442 are used.

Symbol 1452 is a "cage type" or "split case type" of packing similar to that described for symbol 1450 except that rings described under symbol 1443 are used.

Symbol 1600 is a "cup leather" packing of fixed size. It is made of light weight vegetable tanned leather. The tensile strength will be 3000 pounds per square inch. The elongation will be less than 15 per cent in 2 inches with a stress of 2500 pounds per square inch. It will not absorb more than 15 per cent by weight of water.

Symbol 1700 is a "carbon ring" packing of fixed size. It is made of refined coke and/or graphite held together by a suitable binder. The rings are machined to the desired dimensions, the flat sides being parallel and smooth. The inside diameter is ground true and at perfect right angles to the sides. Each ring is made in two or more segments so cut as to permit take up as wear occurs.

The segments are held together by a coiled circumferential spring which lies in a groove on outside rim of the packing.

Symbol 2150 is a "compressed asbestos sheet" packing. It is made of asbestos fiber and rubber compound, two plies or multiples thereof being cross laminated to make the required thickness. In making the separate plies, the asbestos fiber and rubber compound are mixed and felted on a large roll into a sheet of compact and uniform texture with smooth surfaces.

The composition of the packing is as follows:

Asbestos fiber	75 per cent
Rubber	9 to 16 per cent
Sulphur	0.5 to 1.0 per cent
Mineral fillers	13 per cent

The fiber is chrysotile asbestos of not less than 12 per cent water of composition. The rubber is the best quality of new wild or plantation.

The tensile strength is 4000 pounds per square inch at any point on the sheet.

The weight of the finished packing is as follows:

Thickness (Inch)	Pounds per Square Yard	
	Minimum	Maximum
1/64	1.0	1.3
1/32	2.2	2.7
1/16	4.4	5.4
1/8	8.8	10.8
3/16	13.2	16.2
1/4	17.6	21.6

Symbol 2151 is an "asbestos-metallic cloth sheet" packing. It is made of wire insertion woven asbestos cloth treated with rubber compound. The cloth is woven from yarn containing not less than 90 per cent asbestos of not less than 12 per cent water of composition. The weave of the cloth consists of 20 strands in the warp and 10 strands in the filler per linear inch. Each strand

consists of two plies of asbestos yarn and one brass or copper wire twisted together. The brass or copper wire is 0.006 to 0.008 inch in diameter.

The finished packing will weigh as follows:

Thickness	Weight per Square Yard	
	Minimum	Maximum
Inches	Pounds	Pounds
1/16	5.50	6.25
1/8	10.50	11.75
3/16	16.00	17.75
1/4	21.50	24.00

Packing 1/16 inch thick is made of one ply of cloth; 1/8, 3/16 and 1/4 inch thick are made of two, three and four plies respectively of 1/16 inch cloth, cemented together.

This is not more than 50 per cent of rubber compound.

Symbol 2152 is an "asbestos metallic cloth" gasket of fixed size. It is made from wire insertion woven asbestos cloth treated with a rubber compound. The cloth is woven from asbestos yarn and wire of the same composition as described for symbol 2151. The rubber compound will be not less than 40 nor more than 50 per cent. For certain standard sizes of gaskets for handholes, the gasket is seamless, that is, it is folded from a length of cloth woven in cylindrical form. All other sizes are folded from a strip of bias cloth joined at the ends.

Symbol 2153 is an "asbestos metallic cloth" tape made of the same materials and except for the width in the same manner as symbol 2151.

Symbol 2154 is an "asbestos cloth" made without wire insertion or rubber compound. The cloth is woven of asbestos yarn there being 20 strands in the warp and 10 strands in the filler. The asbestos yarn contains 90 per cent of asbestos the water of composition of which is not less than 12 per cent. Cloth 1/16 inch thick weighs 2.25 pounds per square yard.

Symbol 2226 is a "hard fiber sheet" packing. It is made by treating all cotton cellulose paper with zinc chloride. Sizing and

foreign materials other than dyes or pigments for coloring are not used. The finished packing has a clean, hard, straight and smooth surface. Generally the color is red, or natural tan. The specific gravity is 1.15 to 1.50. It will contain not more than 5 per cent mineral matter limited to oxide of iron and clay. The tensile strength for sheets less than 1/16 inch thick is 5000 pounds per square inch and for sheets 1/16 inch to 1/4 inch is 8000 pounds per square inch.

Symbol 2227 is "chart paper" shellacked, and used as a sheet packing. It is made from all rag stock sized with glue, and given a lithograph finish. The thickness will be approximately 0.007 inch. The bursting strength will be 75 points. The ash will not exceed 3 per cent. It will have a folding endurance of 500 double folds.

Symbol 2290 is a "plant fiber sheet" packing. It is made of plant fiber such as jute, hemp, etc., mixed with glue, or other suitable binder, and rolled into the form of a sheet. It is then waterproofed with glycerine or a mixture of glycerine and formaldehyde. It contains no rubber, asbestos or any material affected by oil, gasoline, or water. The moisture content will not exceed 15 per cent. The tensile strength will be 3000 pounds per square inch.

Symbol 2351 is a "rubber sheet" packing. It is made entirely of rubber compound, has a smooth surface and is of uniform thickness. It contains no oil substitute or reclaimed rubber. The compound contains 80 per cent, by volume, of high grade rubber. The packing will have a tensile strength of 2200 pounds per square inch, an ultimate elongation of "2 to 12" inches, and a set of 20 per cent after a set stretch for 10 minutes of "2 to 11" inches. The tensile strength will not decrease more than 40 per cent after subjecting the packing to a dry air temperature of 160 degrees F. for 96 hours. The finished weight of the packing per square yard is tabulated on page 584.

Symbol 2352 is a "rubber-graphite sheet." It is made of rubber compounded with graphite and other mineral fillers and is furnished in an unvulcanized condition.

The compound contains 20 per cent by weight of high grade rubber, 24 to 32 per cent of graphite and about 10 to 15 per cent

Size of Packing	Minimum	Maximum
Inches	Pounds	Pounds
1/32	2.5	3.0
1/16	4.5	5.5
3/32	7.0	8.25
1/8	9.0	11.0
3/16	13.5	16.5
1/4	18.0	22.0

of free sulphur, the balance being mineral fillers. The hardness by plastometer will be 0.40 millimeter. The finished packing 1/32 inch thick will weigh 2.60 to 3.00 pounds per square yard.

Symbol 2353 is a high grade "cloth insertion rubber sheet" packing. It is made of alternate layers of rubber and frictioned cotton sheeting. Both faces of the packing is rubber compound. The rubber compound is properly vulcanized. There is one ply of fabric to each 1/16 inch of packing. The fabric insertion will have 35 to 50 threads per inch in the warp and in the filling and will weigh from 4 3/4 to 5 1/3 ounces per square yard. The rubber layers contain 70 per cent, by volume, of high grade new rubber. The rubber compound will have a tensile strength of 1200 pounds per square inch, an ultimate elongation of "2 to 10" inches, and a set of 25 per cent after a set stretch for 10 minutes of "2 to 8" inches. The weight of the finished packing will be as follows:

Size of Packing	Weight of Packing (Pound per square Yard) Grade I	
Inch	Minimum	Maximum
1/16	4.5	5.5
3/32	7.0	8.0
1/8	9.0	11.0
3/16	13.5	16.0
1/4	18.5	21.5

Symbol 2354 is a "wire insertion rubber sheet" packing. It is made of layers of rubber with brass wire insertion so arranged that both faces is rubber. The rubber layers will contain 60 per cent, by volume, of high grade rubber. The wire insertion contains 24 meshes per inch and is made of wire 0.012 inch in diameter. There is one ply of wire insertion for each 1/16 inch in thickness of packing. The finished packing will weigh as follows:

Thickness of Packing	Weight of Packing per Square Yard	
	Minimum	Maximum
Inches	Pounds	Pounds
1/16	7.0	8.5
3/32	11.0	12.5
1/8	14.0	16.5
3/16	20.5	24.5
1/4	27.0	32.5

Symbol 2410 is a "metallic incased" gasket of fixed size. It is made by incasing asbestos with a copper jacket and is so formed as to surround almost completely the asbestos core. The seam of the jacket or the exposed part of the core will be on one side and not on the edge of the gasket where it may be exposed to gaseous or liquid pressure. The jacket is made of annealed copper not less than 0.008 and not more than 0.015 inch thick. The core is made of soft asbestos millboard or of twisted or braided asbestos cord.

Symbol 2411 is a "metallic incased" gasket of fixed size. It is exactly the same as symbol 2410 except that the jacket is made of monel metal.

Symbol 2412 is a "metallic incased" gasket of fixed size. It is exactly the same as symbol 2410 except that the jacket is made of steel which has been rust proofed.

Symbol 2413 is a "metallic incased" gasket of fixed size. It is exactly the same as symbol 2410 except that the core is made of compressed cork (cork board). The cork will have no binder except the natural cork gum.

Symbol 2470 is a "soft iron" gasket of fixed size. It is cut from soft annealed and pickled sheet iron known commercially as Armco iron. The tensile strength will be about 50,000 pounds per square inch. The carbon will be less than 0.10 per cent. The sulphur and phosphorous will be less than 0.03 per cent. The brinell hardness will be less than 75.

Symbol 2471 is an "aluminum" gasket of fixed size. It is cut from soft annealed sheet aluminum. The tensile strength will be 12,000 pounds per square inch. The elongation in 2 inches will be 15 to 25 per cent depending upon the thickness, the higher elongation being obtained on the thicker sheets. The composition will be 99 per cent aluminum, on analysis.

Symbol 2472 is a "copper" gasket of fixed size. It is cut from soft annealed sheet copper. The tensile strength will be not more than 40,000 pounds per square inch. The elongation in 2 inches will be 15 to 25 per cent depending upon the thickness the higher elongation being obtained on the thicker sheets. The composition will be 99.88 per cent copper, on analysis.

Symbol 2473 is a "lead" gasket of fixed size which is cut from soft sheet lead. The composition will be 99.5 per cent lead, on analysis.

Symbol 2500 is a "leak sealing" compound. It is a rubber-graphite compound furnished in plastic form for application with a pressure gun. Its use also requires a considerable quantity of equipment of a special nature. It should never be used for other than fixed joints.

Table II and the preceding descriptions cover 50 types of packing of which 29 are for rods, shafts and valve stems and 21 are for flanged fittings. This is a greater number than will be required for regular use in a marine power plant. The really active or commonly used packings are symbols 1100, 1101, 1260, 1300, 1301, 1302, 1400, 1430, 1431, 1435, 2150, 2151, 2152, 2290, 2351, 2353 and 2354. The average operating engineer can maintain an efficient plant with the packings mentioned and without any of the so-called special types sold under proprietary designations. By keeping the number of types of packing necessary to obtain satisfactory results at a minimum and using only standard packings there will be quite a saving annually in plant maintenance in addi-

tion to a reduction in the amount of money invested in special packings carried to meet an emergency. This saving may be further increased by standardizing stuffing boxes so that from three to four sizes of rod packing will answer all requirements. In this respect plastic packing in bulk form is the ideal packing material since it can be used in any stuffing box regardless of size.

While it is important to use packing made of the proper material for a given service, it is equally if not more important that its installation and maintenance be properly done. For rod packing all of the old packing should be removed and the gland cleaned. The packing should be installed in rings. These should be cut square, not beveled. For steam and hot liquids, a 1/4 inch opening should be left between the ends of the ring, for all other services the ends of the rings should butt. Each ring should be put well home in the gland, care being taken to see that it does not "cant," until the box is filled. The gland nuts should then be drawn up evenly with a wrench, then slackened back until only finger tight. The packing should be run under this condition for a few hours to give the packing a chance to expand and to get seated on the rod. As the packing expands and leakage disappears it may again be necessary to back on the nuts until finger tight. If these adjustments are made for a few days, the packing will seat itself and need no further attention for months. It will not be necessary later to apply excessive pressure to the packing by means of a long handled wrench to make it appear to be tight. In fact, the application of pressure to the packing by this means should almost never be done for the reason that this creates friction and wears both the packing and the rod. When this means appears to be necessary it is a certain indication of improper installation. Practically any type of packing will work if it has been properly selected, installed and nursed during the wearing in process.

Plastic packing may be installed quite as readily as rod packing, but somewhat more care may be required to prevent waste of the material and to obtain equal density in the box. If the annular clearance between the rod and the throat of the stuffing box or gland is more than 1/32 inch, end rings should be machined and installed so as to reduce the clearance to this limit. As the packing is plastic it can be rolled in the hand like putty into a strip of

sufficient length to make one turn of the rod. This should be placed in the stuffing box and tamped home uniformly with a flat ended stick, repeating the process until the box is completely filled, after which the gland nuts should be drawn up evenly until the packing is tight. It may be necessary to remove the gland and install additional packing before starting the rod or shaft in operation. This packing should also be nursed until it sets to the rod, after which it will run for months without attention.

For flange fittings, the thinnest packing that can be used should be the rule. This will be governed by the alignment, the condition of the flanges and the distance between flanges. Both flange faces should be carefully cleaned of any foreign substance by scraping if necessary. The gasket should also be clean and should be installed without shellac, white lead or similar material. Tighten all bolts gradually and in order, applying pressure equally over the entire flange. Bolts should not be drawn down tight until the third circuit of the flange. Go over the bolts again after the line is put in service. If there is continued gasket trouble with any pair of flanges look for a warped flange or a groove. Flange packing has to be just as carefully installed and nursed as for rods. Do not apply so much pressure as to squeeze the rubber compound out of the packing, as if this is done early failure will result.

NOTES ON CASTINGS.

By Captain L. Shane, U. S. Navy, Member.*

COMPOSITION G.

For some years there has been considerable trouble encountered by various manufacturers as well as the Navy Yards in making Composition G castings. As a rule the physical requirements are met, but it is difficult to get sound castings that will stand hydraulic pressure. Because of this difficulty there has sprung up a tendency to substitute Composition M for Composition G. The use of Composition M for Composition G is inexcusable. The physical properties are so inferior to good Composition G that the substitution should not be considered.

One thing that very few seem to have noticed is that the difficulty in making Composition G castings is almost exactly contemporaneous with the introduction of the open flame furnace, such as the Schwartz, or Rockwell, or Monarch furnaces. And in several cases where this was pointed out to the manufacturers, the fact that these furnaces furnished large capacity in small space, together with the fact that a considerable investment was involved, prevented these manufacturers from listening with a sympathetic ear to the proposed remedy for the troubles.

The writer has solved this trouble in several places, reducing losses to an almost negligible extent by merely requiring the manufacturing plants where he has been on duty to return to practices that were in operation when he entered the service about 35 years ago. There is nothing new in the method proposed except perhaps the requirement that a reliable pyrometer be used. In other words, the remedy is to melt the metal in a crucible where the flames of combustion will not touch the metal, thus preventing burning of the metal, segregation, or any other deleterious action of the flames. Otherwise, merely good foundry practice is all that is required.

It does not matter how the heat is applied, whether by oil or coke fires or by electricity. It is only necessary that the products

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of combustion do not come in contact with the metal. The pyrometer should be used so that there will be no guess work about the temperatures. Many foundrymen do not want to use a pyrometer. They think they can better judge the temperature of metals by eye, but with all due respect to their experience, the eye cannot approach a good pyrometer in telling temperatures. On a dark day dull metal looks bright, and on a bright day hot metal looks dull, so that it is next to impossible to tell within two or three hundred degrees what the actual temperature of the metal is. The pyrometer, of course, solves this problem. With all ingredients fixed, and with the procedure definitely established, it is not difficult to get identical results at all times.

Occasionally a particular type of casting may give trouble. This is an example of mere foundry troubles which must be rectified by a careful study of the problem. Sometimes it is a matter of gating, other times the section of the casting may be so varying as to make it almost impossible to make the casting without porous spots. This can be rectified in several ways, such as slight alteration of the pattern to give a more uniform section, or perhaps if this is not practicable, by the use of chills. However, these are troubles that the good foundryman will solve, and are not inherent in bronze castings, but common to all kinds of foundry work, and we are not at this time considering anything but Composition G.

The following is the procedure in effect at the Portsmouth Navy Yard, where the difficulties in the manufacture of Composition G have been entirely overcome.

(*a*) Metal is melted in coke-fired crucibles for all castings requiring hydraulic tests, and where uniform homogeneity is essential. Pouring temperature is 2300 degrees F.

(Either pit crucibles, or tilting crucibles, oil-fired, will do).

(*b*) Metal is put into the crucible in the following sequence:

(1) Gates and sprues of Composition G melts, and copper.
(2) When this metal is melted and hot
(3) Zinc is next added and stirred in well
(4) Tin then is added and
(5) When the crucible is pulled out, about two (2) ounces of phosphor copper to 100 pounds of metal is added and stirred.

June 1933 Capt Shane says item 5 should be deleted.
HBA

(*c*) In melting keep the metal well covered with charcoal.

(*d*) In molding Composition G, firm, clean molds are necessary. Long runners are not used and the sprue is kept close to the casting where it is gated. Risers are moderate in size and numbers; that is, there are not as many nor are they as large as for the brasses.

The machining of the Portsmouth Composition G castings indicates great toughness, and the tightness tests show that the material is as dense as it is strong. For example, the following physical characteristics have recently been obtained on high pressure valve bodies:

Test specimen	*Yield point*	*Tensile strength*	*Elongation per cent*
G-59	20,450	49,700	47
G-60	21,500	46,350	35

A comparison of fractures of two specimens, one of crucible brass, and the other of brass produced in an open flame furnace, will show the difference. The crucible specimen will show a perfectly homogeneous fracture, with no discoloration, while the open flame specimen will show much segregation, and many black or green particles.

The writer is offering nothing new in this article, but is merely recommending a return to methods that were in successful use 35 years ago. Many officers were not in the Navy at that time, and it has been so long since uniformly good castings of Composition G have been made, that the vast majority of officers are not familiar with practices of so long ago. It is with the idea of spreading this information throughout the service that this article is written.

MANGANESE BRONZE.

This is a metal which is in constant use in the service where a brass of high physical qualities is needed together with high elongation and toughness. However, there are as many different kinds of manganese bronze as there are manufacturers that make it, and no two are alike. The physical qualities required are as follows:

Tensile strength .....65,000 pounds
Elongation .....20 per cent in two inches

As a rule the cost of the ingot for castings in the open market runs about 22 cents a pound, a rather high price for a low cost to produce product. For the forging bronze the price runs from 40 to 50 cents a pound.

This material is usually purchased at nearly all the Navy Yards. It is similar to vanadium bronze and to titanium bronze, the principal difference being in the choice of a scavenger, whether ferro-manganese, ferro-vanadium, or ferro-titanium, the amount of any of these scavengers varying with the individual experiences of the various producers of the metal. Strictly speaking, the modern manganese bronze is not a bronze at all. The element of tin has been left out and the principal ingredients are copper and zinc, so that the metal should by rights be called manganese brass.

About the year 1908 there was developed at the Philadelphia Navy Yard a manganese bronze which later was given the designation by the Navy Department of " Navy Composition C." It was the result of about a year and a half of experimentation by Mr. L. A. Kenney, chief draughtsman, Mr. Thomas Mayer, master foundryman, and the writer, and was originally based on a formula with which Mr. Mayer was familiar. This formula was modified in various ways, by the inclusion of various amounts of tin as well as of copper, zinc, iron and ferro-manganese. One formula produced a bronze of a tensile strength of 120,000 pounds, but as brittle as cast iron, and not nearly as useful. Eventually a formula was evolved which had high physical characteristics, and yet was finer for forging than any of the regular forging bronzes in the market.

The Philadelphia Navy Yard produced this bronze for years (and perhaps is still doing so), and the writer has seen it described in a textbook on foundry practice, and called " Manganese Bronze, Philadelphia Navy Yard Type." And yet in no other Navy Yard that the writer is familiar with is this bronze being produced, with the exception of the Portsmouth Navy Yard where he now is on duty and where he introduced it. It can be produced for about 12 cents a pound depending on the accounting system, and, as before said, is a finer bronze for forging at 12 cents than the best obtainable in the market at 40 cents or more.

In order that this material may be brought to the attention of all the service with a view to making full use of one of the most valuable materials used in construction, at a low cost, the process of making this bronze will be described.

The procedure must be followed exactly. The greatest difficulty in teaching the average foundryman to make this metal is to get him to follow instructions implicitly, without injecting some of his own ideas into the process. Where the method is followed exactly, identical results to that obtained at Philadelphia will be obtained.

The following are the physical characteristics of this bronze:

Tensile strength	86,500
Yield point	46,500
Elongation	25.8 per cent in two inches

The following are the ingredients per 100 pounds:

Copper	56
Zinc	37
Ferro-manganese	4
Iron (Norway)	1¾
Aluminum	1¼

METHOD OF MANUFACTURE.

First, melt the copper, iron, and ferro-manganese in a crucible, heating until very hot and fluid.

Second, stir in the zinc.

Third, stir in the aluminum.

Stir well, then pour into pigs.

For casting, use the pigs, for the results are obtained on the second melting.

The aluminum is used to help the fluidity of the metal.

In making castings only the usual good foundry practice is needed, but both for the original ingots, as well as for the castings, the metal must be melted in crucibles. In this connection, when making castings care should be taken to melt the metal down expeditiously. It should not take more than 1¾ hours, and certainly not more than two hours. If too long a time is taken, much of the zinc is burned out, and the ultimate strength and yield point

are reduced, while the elongation rises. If as much as four hours are taken, the ultimate strength may fall to a little over 70,000 pounds, the yield point to 33,000 pounds, but the elongation may rise to 45 to 50 per cent. However, the forging qualities are always excellent. This has not been impaired even if the time of melting is over long.

Forging has no effect whatever on the physical properties of this metal. A bar five inches in diameter can be heated to a cherry red, and forged down to almost any diameter, 1/16 inch or less, without re-heating. This statement sounds extravagant, but it is literally true.

In making the casting the pyrometer should be in evidence. The temperature of pouring should be as nearly as possible 2100 degrees F. This is considerably higher than is usually advocated, but it is essential that the metal be fluid and shall not set, nor tend to set, before the mold is filled. Generous gates and risers should be provided, and they should be churned as is usual in manganese castings, for this metal has high shrinkage and the gates must be large enough to feed the casting, and not the casting feed the gates.

If the foregoing technique is closely followed, there will be very little trouble getting excellent manganese bronze castings.

For forging, make a mold of the most convenient size for the forgeman and forge in the usual manner. There is no difficulty in forging this metal, in fact it forges more easily than steel, for there is no critical temperature to worry the operator.

MONEL CASTING.

Experience has demonstrated that most foundries have considerable trouble in making monel metal castings. At least, it is difficult for them to produce castings that are not porous. During the construction of the Scout Cruisers *Omaha, Milwaukee,* and *Cincinnati,* by the Todd Drydock and Construction Corporation at Tacoma, Washington, many of the high pressure steam valves were designed with monel metal valve seats. It was practically impossible to obtain cast valve seats that would not leak under hydraulic pressure. In fact, more than 95 per cent of the castings furnished by the contractor who was the representative of the

International Nickel Company, and who at least should have known how to handle the metal for which he was the agent, were rejected. The situation became so serious that the contract was finally canceled. It was then decided to attempt making these castings in the Todd foundry, buying the monel shot from the representative of the International Nickel Company.

After some experimenting, a method was finally evolved which produced about 90 per cent good castings. This method will now be described.

An oil-fired tilting crucible furnace was used. In this regard, it is absolutely essential that the metal be melted in a crucible to avoid the introduction of deleterious substances when the flame is in contact with the metal, thus tending to porosity in the casting. It is also highly important that the pot be kept covered during the melting process.

Charge the required amount of monel metal in a covered crucible, using (sixty) 60 per cent new metal and 40 per cent scrap. Use borax as a flux. Keep cover on the pot and heat to twenty-eight hundred fifty degrees (2850 degrees) F. There will be more or less slag formed at the top of the metal when the charge is melted. On removing the crucible from the furnace this slag should tend to collect at the side of the crucible, the metal showing in the center of the crucible, with iridescent colors moving to the sides of the crucible. (Use dark glasses when looking at the metal.)

Just before pouring, introduce one ounce of metallic magnesium per 100 pounds of metal and stir vigorously, using an iron rod to introduce the magnesium. Keep the cover over the top as much as possible to protect the melter, as the introduction of the magnesium is accompanied by considerable sputtering and flame.

Do not skim off the slag, but pour direct from the crucible in which the metal was melted, pushing the slag back with a skimmer.

In the proper kind of tilting furnace the metal can be poured into the molds direct.

It will be apparent that the time element is vital in making a successful pour. It all resolves itself into a question of temperatures. The temperature must be high, in order to keep the metal sufficiently fluid so that no setting occurs before the mold is filled.

Pour rapidly using large gates—(Use pop gates for valve seats). Chill all corners of the mold that tend to draw, as monel castings cannot be fed up as with manganese bronze.

In preparing the mold use seven parts of silica sand, one part fire clay, one part used molding sand, and one part fine sawdust. Wet down with molasses water and mix in a mill. Ram the sand as for steel castings, and dry well in the oven.

Line the crucible with a mixture of 50 per cent Johns-Manville No. 31 and 50 per cent silica sand. This adds to the lasting qualities of the pot.

It is considered better practice to carry the mold to the furnace and pour direct from the pot; this, in order to avoid chilling the metal while removing pot from the furnace.

A NEW CRITERION OF CAVITATION.

By Comdr. James M. Irish, U. S. Navy, Member.

When the phenomenon of propeller action called "cavitation" was first noted and discussed by Barnaby in connection with the *Daring's* trials, there was opened up a field of discussion which has a direct bearing on every theory of propeller design. It immediately pointed to a condition which must be avoided in developing a particular design, and it offered a possible explanation for the failure of certain propellers to equal predicted results. Since cavitation is a possibility, every theory of propeller design which pretends to be complete must offer means for determining whether a particular propeller operating under given conditions is or is not in this dangerous area; and, furthermore, this method must be applicable not only to the design of a propeller but also to the results obtained on actual trials. Thus a correct method must show that a propeller which is grossly unsatisfactory is in cavitation, but also that an excellent performance is due to its being not in cavitation.

The fascination which surrounds the study of propeller action is increased by the introduction of cavitation because, except in rare cases, it is impossible definitely to say that cavitation is present. Conditions which cause cavitation in actual ships' propellers cannot be duplicated in the model tank, nor can the effects on actual ships be shown except by comparison with theoretical results, or by extensive comparative trials with various propellers. These indeterminates have therefore resulted in wide differences of opinion, dependent on the theory adopted. In the model tank for high speed ships, such as destroyers and cruisers, it is assumed that cavitation is present after reaching speeds of approximately 28 knots, and above this point the actual S.H.P. and R.P.M. arrived at from test of model screw and hull are increased by

certain factors to take care of the effect of this cavitation. Yet on the same hull and propeller, by means of Dyson's method, correct results, checking closely with trial figures, are obtained in R.P.M. and S.H.P., *making no allowance for cavitation.**

On the other hand cavitation is assumed when it is doubtful if any really exists. In certain sister ships with practically identical propellers, we find Admiral Dyson† analyzing Ship No. 1 as being in cavitation which has a propulsive coefficient (based on S.H.P.) of .734, whereas No. 6 has a propulsive coefficient of .692 yet is said by the author to be *not* in cavitation. If the propeller of Ship No. 1 is in cavitation then the effects are certainly not very serious.

It would appear therefore that in an endeavor to justify their theory, designers have extended the term "cavitation" to cases where their theory is more probably at fault.

On the other hand in certain outstanding cases, like the *Daring*, the *Drake* or the *Perry*, true cavitation did actually exist and the performance of the ship was noticeably improved by changes in the propellers. These examples were reviewed and the data given in an excellent article in "Shipbuilding and Shipping Record" for March 21, 1925, "The Criterion of Cavitation." There is another group of trials in which cavitation undoubtedly exists and which are useful in studying its results, namely, the submarines. Here is found a direct comparison, with the same screw, between normal surface operation and submerged condition with high thrusts.

These results compared in Table I show a remarkable resemblance to those obtained on the *Daring's* trials, and from them two conclusions may be drawn: First, that there are limits to the load within which a propeller will operate normally, but outside of which there is a radical change in efficiency, and second, that outside these limits the propeller operates at a greatly increased apparent slip. In fact these phenomena lead to the conclusion that when the load per revolution is too great, the propeller tends to correct this by automatically increasing its speed so as to carry this load.

* See Problem 40, 3rd Edition of Dyson's "Screw Propellers."
† See Problem 6, 3rd Edition of Dyson's "Screw Propellers."

TABLE I.

Daring.

Trial No. 1 (Cavitating)			Trial No. 6		
V	I.H.P.	R.P.M.	V	I.H.P.	R.P.M.
16	900	235	16	750	200
18	1300	260	18	1150	235
20	2000	300	20	1750	265
22	2750	340	22	2400	300
24	3700	385	24	3000	328
26	.....	.....	26	3700	355
28	.....	.....	28	4400	385

Submarine. (Prob. 6, Dyson 3d Ed., page 288)

Submerged (Cavitating)				Surface			
V	S.H.P.	R.P.M.	E.H.P.	V	S.H.P.	R.P.M.	E.H.P.
6.93	272	175	177	7	140	143	100
7.81	388	200	236	8	200	160	144
10.12	860	257	472	10	370	203	268
11.8	1380	297	709	12	632	247	444
13.12	1840	332	945	14	1134	300	740
				16	1670	345	1050

Numerous investigators have suggested methods for determining the limit within which cavitation need not be expected, and these results are investigated and applied in the article in "Shipbuilding and Shipping Record," referred to above. The conclusions reached in this article are "that the pressure per square inch method suggested by Barnaby and supported in a modified way by Normand and Dyson affords a satisfactory method of indicating the zone in which cavitation may be feared."

Barnaby in his 1911 paper to the Institute of Naval Architects gives a limit of thrust of 13 pounds per square inch. Normand's in 1902 summed up his conclusions in the equation

$$N D^2 \gamma^{3/4} = C \frac{P}{V^2}$$

where

N = number of screws

D = diameter of propeller in meters

$\gamma = \frac{\text{developed area}}{\text{disc area}}$

P = maximum horsepower (I.H.P.) (Total)

V = corresponding speed of ship in knots

C is a factor which should not be less than 0.6 and can with advantage exceed 0.8

It will be noted that neither of these limits give any value to revolutions per minute, or to its equivalent, tip speed, and, as more recent experience has demonstrated, the pressure limit can be raised as the revolutions are increased. One other feature which will be referred to again is Normand's use of power transmitted to the propeller as compared with other investigators' use of the pressure calculated from the effective thrust. This is unique and valuable in that it avoids the use of the thrust deduction for determining thrust horsepower which is an unknown and variable quantity.

Admiral Dyson's method of determining cavitation is undoubtedly the most complete of any and the varying factors of pressure and tip speed are fully allowed for, but to apply his test to a given propeller it is necessary to use his system in its entirety. To one not thoroughly acquainted with the Dyson system, this is impossible.

The author has, therefore, endeavored to evolve a criterion for cavitation which will embody the essentials as demonstrated by cases of actual cavitation, but which will be independent of any particular method of design. A study of the results obtained on a large number of ship trials pointed to the conclusion that the shaft horsepower which could be absorbed and transmitted efficiently by a given propeller depended on two things: (*a*) the area of the propeller, and (*b*) the tip speed. It is therefore possible

to plot these on rectangular co-ordinates by expressing the pressure in terms of the unit area:

$$\text{Pressure per square inch} = \frac{\text{SHP} \times 326}{\text{N} \times \text{V}} \div \text{Pa}$$

where S.H.P. = total shaft horsepower.
N = Number of shafts.
V = speed of vessel in knots
Pa = Projected area in square inches
this value is hereafter expressed as T/□″
and Tip Speed = $\pi \times D \times R$
where D = diameter of propeller in feet
and R = revolutions per minute

It will be noted that the pressure is calculated from the S.H.P. rather than the E.H.P., which is the basis used by M. Normand. As stated above this is believed to be more correct as it avoids the use of the unknown quantity, thrust deduction.

By plotting a large number of vessels whose poor performance indicated cavitation was present, as compared with other vessels with highly satisfactory results, a curve of limits was finally developed which demonstrated that whenever a propeller was designed so as to operate below these limits normal efficiencies and revolutions would be obtained. This curve is given in Sheet I.

To illustrate, the data is plotted for the U.S.S. *New Mexico* which was standardized with four shafts and again with two shafts.*

Ship Data	Propeller Data
L. W. L. 600′	Number 4
B 97′ 4½″	Pa/Da .3567
H 30′	Dia. 13.412
Displacement 32000 Tons	P 15.167
N. B. C. .635	
Coefficient M. S. .981	

* A.S.N.E. Journal, May and August, 1919.

	4 Shaft Trial				2 Shaft Trial		
V	21	19.0	15	10	15	13.5	10
Act. SHP (Total)	29100	20800	9175	3200	11586	9263	4443
R.P.M. (Ave.)	166.7	149.2	115.6	78.4	143.85	132.0	101.9
E.H.P. (Tank) (Total)	20550	14890	6780	2333	6780	5160	2333
E.H.P.*	....	....	....	....	*7600	5790	2610
P.C.	.705	.715	.738	.728	.654	.626	.532
Pa = 7230 Square Inches.							
T.S.	7040	6280	4860	3300	6040	5560	4280
T	112800	89300	49800	26000	126000	111800	72200
T/□″	15.60	12.35	6.9	3.61	17.4	15.5	10.0

It will be seen that when the pressure(T/□″)is plotted against tip speed that the points for the 4-shaft trial are below the cavitation limit, the 21-knot point being practically on the line, whereas for the 2-shaft trial the points are above the limit. The question now arises how the designer would know prior to the 2-shaft trial that cavitation would exist. If we assume normal performance for 15 knots the estimated S.H.P. would be 10,600 and the R.P.M. 137.0. (These figures were arrived at by using the Dyson method without cavitation.) The corresponding tip speed is 5760 with a T/□″of 15.9; as the limiting pressure for 5760 is 11.8 it is evident that cavitation is to be expected. This example will also show the fallacy of the fixed limit of pressure based on the effective thrust. This thrust equals $\frac{7600 \times 326}{2 \times 15} = 82{,}500$ which corresponds to a pressure per square inch of 11.4 pounds. A designer, using the 13 pounds limit established as safe by Barnaby, would have been at a loss to fix the reason for this lack of agreement between estimate and result. Checking the results by Normand's equation shows a value of "C" for the 2-shaft trial of 0.36 which would indicate cavitation; but for the 4-shaft trial the value of "C" is 0.51 which is below the allowed limit and would lead the designer to expect cavitation, whereas on trial a propulsive coeffi-

* 12 per cent increase is allowed for two dragging screws.

cient of better than .70 was obtained, so that cavitation was not serious if it existed at all.

Another example of cavitation corrected by use of proper area is given in the article by Slocum in "Marine Engineering and Shipping Age" for February 1926. This ship, the *Challenger,* was a single screw vessel on which three different propellers were tried on three separate voyages. The following table gives the reported results:

Ship		Propeller No. 1	Propeller No. 2	Propeller No. 3
Data	No. of shafts	1	1	1
L 410	No. of blades	4	4	4
B 56	Pa/Da (total)	.368	.373	.432
H 30.6	D	17.0	17.0	17.0
Displ. 15930	P	16.5	16.0	14.075
⊗ .99	Dev. Area	98 □′	98 □′	111 □′
N. B. C. .795	Air speed	9.63	10.4	10.83
	R. P. M.	70.12	75.92	82
	S. H. P.	2117	2501	2537
Tank	E. H. P.	965	1223	1373
	P. C.	.445	.49	.542
	T. S.	3750	4150	4380
	T	71500	78400	76600
	Pa	12000 □″	12200	14100
	T/□″	5.96	6.41	5.43
Cav.	limit	5.0	6.20	7.00

This ship offers an excellent illustration of the effect of too little propeller area, and of the gain due to increase in revolutions. The designer of the original propeller no doubt expected a speed of 9.63 knots on approximately 1800 S.H.P. and 68 R.P.M. These could be attained if no cavitation existed. This gives aT/□″ of 5.06 with a tip speed of 3640. From the chart he could have foreseen that cavitation would be present.

By reducing the pitch on the second trip an increase in R.P.M. and tip speed resulted, and it was possible to carry a much larger thrust on the same propeller with an increase in efficiency. Worked

without cavitation, this propeller would have attained 10.4 knots on 2250 S.H.P. at 74.4 R.P.M., corresponding to a T/□″ of 5.78 and a tip speed 4050. This is close to the line, the thrust limit for tip speed of 4050 being 5.70.

Having demonstrated how the chart is to be used, it remains only to show how effects of cavitation on revolutions and power may be estimated. As stated above the tendency for any propeller when a pressure larger than the cavitation limit is being impressed on it is to increase the revolutions.

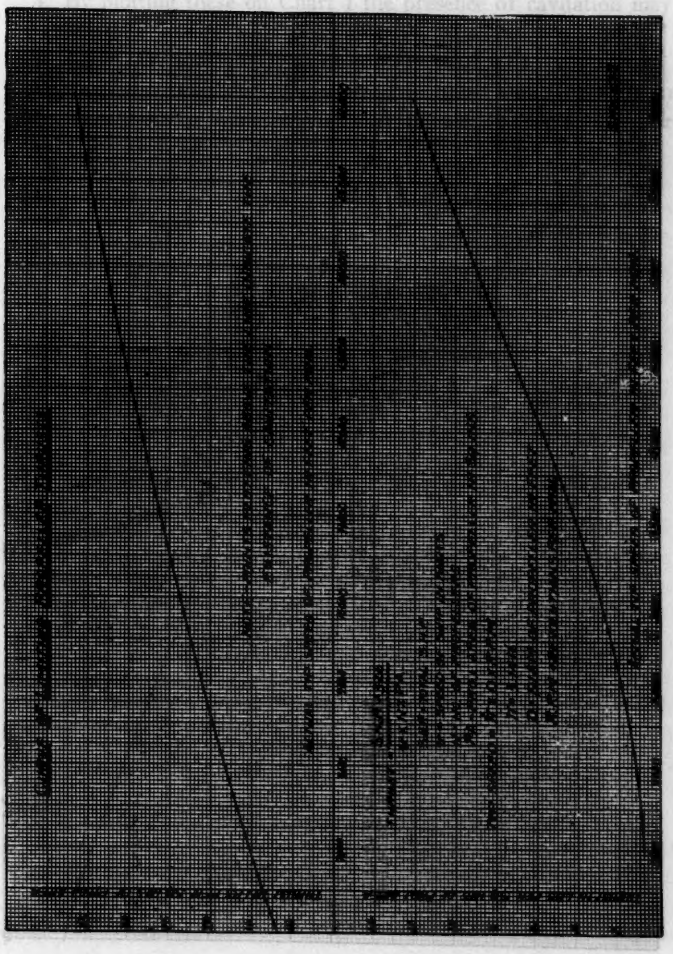
The amount that the pressure exceeds the limit for the estimated tip speed is a measure of the increase. For reasonable limits, within 30 per cent the increase in S.H.P. is 10 per cent and the increase in R.P.M. 5 per cent. Thus for the *Challenger's* first trip the estimated S.H.P. (without cavitation) of 1800 will be increased by 10 per cent to 1970 and the R.P.M. by 5 per cent to 71.6, the actual 2117 S.H.P. and 70.13. For the *New Mexico* the S.H.P. when cavitating is $1.10 \times 10{,}600$ or 11,650, and the R.P.M. 1.05×137 or 144. The actual were 11,586 S.H.P. and 143.85 R.P.M. Above these limits the propeller actually breaks down and the R.P.M. increase very rapidly and no accurate estimate is possible.

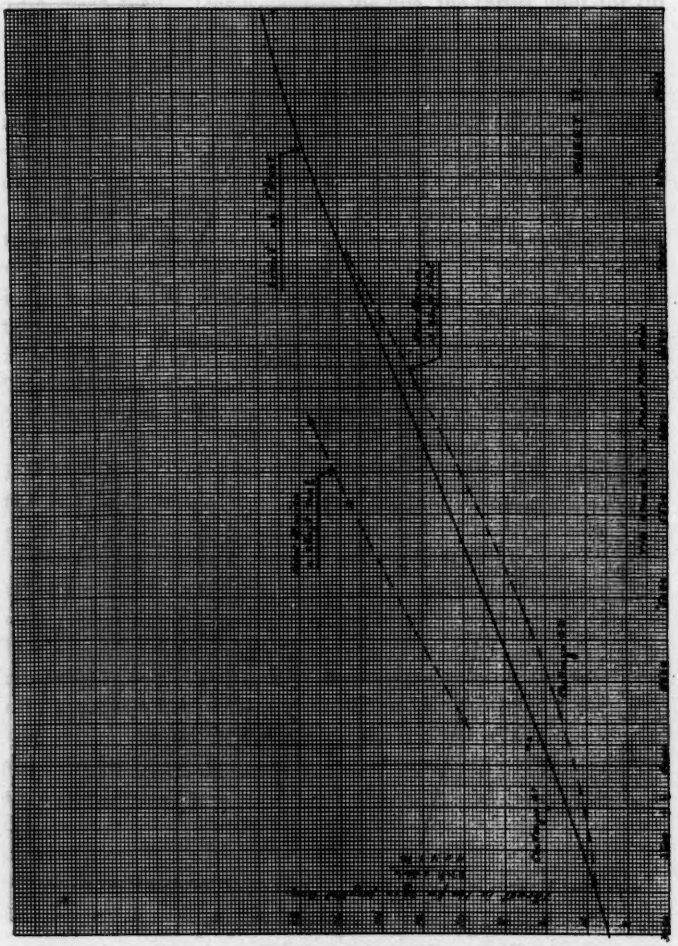
The results of plotting numerous vessels indicate that this cavitation limit is a zone rather than a line, due probably to errors in data. The *New Mexico* for 21 knots plots virtually on the line but shows no decrease in efficiency. For safety it is therefore advisable to use 95 per cent or even 90 per cent of the value read off the curve when using it to fix the proper projected area. This will render the screw safe when the resistance due to adverse winds or fouling is increased over trial conditions. Too much area may counteract the effect for, as shown by Admiral Dyson, the P.C. falls off with increase in projected area ratio.

CONCLUSIONS.

1. Methods of propeller design which neglect cavitation are faulty and are the cause of failure of certain propellers.

2. Present methods of determining cavitation limits which disregard effect of revolutions per minute are misleading.





3. The determining factors are the pressure based on S.H.P. and tip speed.

4. By plotting these on Chart I the presence of cavitation may be foretold.

5. The cavitation limit is a zone rather than a line.

6. Cavitation may be avoided by providing sufficient area to carry the thrust, or by decreasing pitch to raise the revolutions per minute, or by both.

BOILER ROOM OPERATION ON NAVAL VESSELS.

By Commander H. G. Donald, U. S. Navy, Member.

Reliability is certainly the first requisite in the operation of a machinery plant on a Naval vessel, although no one can be considered successful as an operating engineer in the United States Navy of today who cannot also obtain high efficiency as judged by comparative fuel performance in competition with sister type vessels. When one considers the wide scope of the war theater in which our fleet must engage, it is realized that ability to secure economy of operation at all powers is a very close second in importance to reliability. All those engineers who have been successful in attaining this high efficiency will tell you that the game is played and won or lost in the fire room. The fire room, not being the center from which the plant is controlled, is less frequently visited by the Engineer Officer when underway. Fire rooms are relatively inaccessible, hot, shut-in spaces (using forced draft). Control is usually turned over to a chief petty officer or at best a junior officer. These facts tend toward looseness in operation in the fire rooms, and it requires eternal vigilance to cope with these handicaps. Without discounting the importance of efficiency in the operation of the engine room machinery, it is certain that skill in operation and good engineering judgment can, with the same effort, accomplish many times more in the boiler rooms than in the engine rooms.

This is because there are many more variables in fire room operation dependent on the operating personnel. The following is a discussion of these variables in an effort to produce a picture decipherable by the beginner in practical engineering on naval fighting vessels. Stress is laid on fighting vessels for these are operated

Editor's Note.—Many of the suggestions of this article are covered specifically by instructions in the Bureau of Engineering Manual, etc., which should be referred to.

under much more difficult circumstances than are the auxiliary vessels. The operation of the latter is very similar to the operation of merchant vessels, the power used underway being nearly constant and usually employing the entire boiler plant.

This discussion deals with all components of the boiler room, including boilers, burners, blowers, feed water and oil pumping and heating apparatus, and their best coordination in order to obtain maximum efficiency from the boiler room end of the operating problem. In order to obtain the highest efficiency of the combined units, it will in certain cases be advisable to operate some units at other than best conditions because of an otherwise adverse effect on other units having more weight in the heat balance.

As an example, the most efficient operation of a burner is usually at a high point in its capacity range. But if a boiler is being operated at a low rate, the overall fire room efficiency will be adversely affected if the burners are operated at this (high) best rate. This is because the increased steam consumption of the forced draft blowers at the higher fire room air pressure required for the higher burner rate will overbalance the gain in efficiency of combustion due to the use of the (high) best burner rate. It must be remembered that no one can give set formulae for operation covering all of the very varied conditions of operation met with in service with the fleet or fitting all of the assortments of boiler room equipment found on Naval vessels. Judgment backed by experience must always be the guide.

The three major factors entering into efficiency of boiler room operation are:

Number of boilers used.
Rate of combustion used in each boiler.
Rate of combustion used for each burner.

The number of boilers used depends on —

Speed of ship.
Total number of boilers installed.
Number of boilers per fire room.
And a consideration, but not necessarily a deciding point, of the best rate of combustion per boiler.

Full power, of course, involves all boilers, but at speeds less than full power, the usual condition, there is considerable latitude in the

choice of the number of boilers to use. During maneuvers, there should always be a sufficient number of boilers in use to meet the "full" speed required by the conditions of ship operation without burning oil at a higher rate per boiler than about three-quarters of that rate burned per boiler at full power of the ship. Fighting vessels equipped with four or less number of boilers should never steam in company with less than two boilers in use. Those with more than four boilers should normally not use less than one-third the total number installed. The number of fire rooms in use for cruising conditions should not be more than will permit of a watch in four for the men and also provide a working gang for overhaul in idle fire rooms. All boilers in a steaming fire room should be used, as the loss of air through dead boilers outweighs any other possible consideration.

Large vessels entering a crowded or constricted harbor should assure of ample boiler power for emergency backing, and this is usually in excess of the number of boilers employed for low cruising speeds.

The above conditions being satisfied it is well to endeavor to operate the boilers at their best rate of combustion.

The best rate of combustion is, for water tube boilers, between ¼ and ⅓ pound of fuel per square foot of heating surface. The rate of combustion of all boilers used at one time should be as nearly equal as possible. When employing more than one fire room, and where the ship speed is fairly constant it is, however, advisable to employ one of the fire rooms to meet minor fluctuations in steam demand, *i.e.* to maintain main steam line pressure, and set the other fire rooms on a steady unvarying rate of combustion approximating their share of the average steam demand.

The rate of combustion to be used per burner will vary with the type of burner, the number of burners installed per boiler and their location on the boiler front. It will be limited somewhat by the available assortment of atomizer sprayer plates (or tips and plugs for Bureau and Fore River burners). Never mix sprayer plate (tip and plug) sizes on burners in use in one fire room. The fire room air pressure maintained must serve all burners in use and throttling registers to suit a mixture of sprayer plates is not practicable of accomplishment without loss in efficiency. The pos-

sible maximum capacities of types of fuel oil burners vary roughly, directly with the diameter of the opening in the furnace front plate. The maximum capacity of a given type will vary directly with the fire room air pressure available and for a given air pressure will vary inversely with the number of burners installed on a boiler. The principal burners used in the United States Navy, their accepted maximum capacities (at about 8 inches of water fire room air pressure) and best rates of combustion are given in the following table:

Type	Maximum capacity gallons per burner per hour	Best rate (approximate) gallons per burner per hour
Bureau (conical register)	110	45
Fore River (conical register)	93	38
Cuyama (B. and W. Co.)	133	100
Lodi (B. and W. Co.)	230	150

It is best to hold the operating rate of a burner approximately at the best rate within the limitations of the atomizer sprayer plates (or tips and plugs) available, provided the required rate of combustion of the boiler unit does not demand the use of burners set on the boiler front in locations where either the oil spray is apt to strike the refractory lining of the furnace (side walls and floors) before complete gassification or the flame has a tendency to impinge on the tubes. This does not necessarily mean that the peak or bottom row burners should not be used until absolutely required regardless of desired limitations in rate of combustion per burner. In later installations all burners can safely be used, and many older installations having burners disadvantageously located have been corrected by a carefully considered plan of tilting the burners on the boiler front. This tilting should not be attempted by the novice, for if improperly done it may cause other troubles greater than those which it was designed to correct. That burner installation is the best which permits the free use of all burners. In this connection it is of importance that those burners in use be equally distributed to right and left of the center line of the furnace. On a given installation it is well to give the burners serial numbers

which will indicate to the fireman the order in which additional burners will be cut in. In this respect the reasoning of the fireman should not be depended upon.

In considering the rate of combustion per burner to be employed, it may be stated as an axiom that for a given rate of evaporation per boiler, efficiency of the fire room as a whole is improved by using a large number of burners at a low rate per burner rather than a small number of burners at a high rate per burner. The reason lies in the reduced fire room air pressure and consequent increase in that major factor, blower efficiency, as well as in reduced leakage of air into the boiler casing and through leaks in the fire room doors, etc. The firemen dislike to follow this procedure, for not only does lowered fire room pressure usually mean a hotter fire room, but also more of the burners must be handled when meeting changes in steam demand. The way to win the cooperation of the firemen is to instill in them interest in fuel economy and drill them in changing burners just as shellmen are drilled on minor caliber guns. Care must be exercised when oil is secured on a burner, that the atomizer is immediately removel from the boiler. Of course, when the engines are being handled on bell signals, time does not permit removal of the atomizers. But, should an atomizer be left in place, with the oil secured, for an appreciable time, the oil in the tip will stew and partly vaporize, gumming up the small orifices and causing poor atomization when the oil is again turned on.

Having discussed the more general factors of operation governing efficiency in the boiler rooms, consideration will now be given to those of a more detail nature. These may be listed as follows:

Combustion as affected by atomization, mixing of air with oil spray and air supply.

Feeding, effect of variations in rate and temperature.

Steam pressure under various conditions of operation.

COMBUSTION.

ATOMIZATION: In the combustion of fuel oil the first requisite to efficiency is to secure as close to ideal fineness of atomization as may be practicable. It is a point most often neglected. Poor atomization is the cause of many troubles, and often the source of

these troubles is not suspected by operating personnel. Tests for fineness of atomization cannot be properly made on board ship, and certainly not when a boiler is being fired, except in a rough way by noting whether or not abnormal air pressure is required. If this check is used one should be careful to compare the current condition with a standard which used the same identical burners, the same size sprayer plate (or tip and plug) and the same pressure on the oil line at the boiler front.

It is essential to obtaining a satisfactory fineness of atomization that the following points be given attention continuously:

Oil Pressure: This should be the pressure at the boiler front, not at the pump, for the latter pressure may be misleading due to valves between the pump and burners not being fully opened, and, in this connection, the oil valves at the burners should never be throttled. Satisfactory atomization is not obtained at less pressure at the atomizer than 125 pounds gauge. Always use a sprayer plate (or tip and plug) size which will give the burner capacity desired at a pressure between 125 and 300 pounds gauge. It is necessary that the pressure be steady—a 5-pound fluctuation being considered unsatisfactory. Steady pressure is obtained by keeping the service pump in good running order and the air chambers charged.

Viscosity: The proper viscosity recommended by the Fuel Oil Testing Plant is four degrees on the Engler scale. There is a question whether the improved fineness of atomization to be obtained at lower (thinner) viscosity repays the additional heating required. The viscosity-temperature characteristics of fuel oils are such that at the low viscosities they are sluggish to change in viscosity, *i.e.*, a relatively large temperature change is necessary to obtain a small viscosity change. This is fortunate for operation and permits small fluctuations in maintained temperature of the oil without seriously affecting the atomization. However, the temperature required to reduce the viscosity of fuel oils furnished the Navy to the required figure may be anywhere between about 80 degrees F. and 250 degrees F. Usually the higher the oil appears in the Baume scale the lower will be the temperature required to reduce the viscosity to that satisfactory for atomization. The viscosity-temperature characteristics of fuel oils are, however, so

complex that the proper atomizing temperature must be known for each bunkering of oil. In so far as is practicable, keep the different bunkerings separate and burn up odd lots in port. Small ships do not have viscosimeters and large ships having them do not always use them. It is satisfactory, however, to ascertain the atomizing temperature of an oil by use of the viscosity-temperature chart furnished each ship by the Bureau of Engineering and which was devised by Lieutenant Commander G. B. Vroom. To use this chart requires only that one point on the viscosity-temperature curve of a given oil be known. Data for one point are required to be furnished by the tanker and the information should always be obtained before shoving off from the tanker.

The use of a temperature much higher than that required for satisfactory atomization involves danger of the oil reaching the point of gassification in the heater. Such a condition causes first a loss in atomizing capacity and if carried further causes bad panting of the burners. Too low a temperature, the most common fault, is not distinguishable in operation, because unless carried very far (say above 20 degrees Engler in viscosity) it can be corrected by the use of additional excess air to control the smoke. Therein lies the usual main loss in efficiency. In fact, it may be said that most of the faults in combustion can be covered up by raising the quantity of excess air furnished.

It is also a common fault to fail to tend the steam to the fuel oil heater, whenever the rate of combustion is increased until after the oil rate is increased. This results in bad smoking, an admonition from the bridge, and although by the time the word that smoke is excessive reaches the fire room, the oil temperature may have been adjusted, yet the air pressure is run up considerably in excess of requirements and probably held there until the next change in speed. In the meanwhile, economy has been sacrificed. Automatic temperature control devices are furnished some of the capital ships.

Carbon deposit on tubes or refractory is a sure indication of poor atomization unless it is known to be due to poor location of burners.

Wear of Atomizer Sprayer Plates (tips and plugs): Very slight wear of the orifice or grooves leading to the whirling chamber

seriously impairs atomization. Assembly of the sprayer plate (or tip and plug) on the atomizer pipe may be the cause of faults similar to those caused by wear. There must be no leakage of oil past the metal to metal contact surfaces in the sprayer plate (tip and plug) assembly. Burred edges of the grooves leading to the whirling chamber are the most frequent cause of improper assembly.

MIXING AIR AND OIL SPRAY.—*Spray angle:* Although the choice of sprayer plates (or tips and plugs) is usually limited, it is well to know that sprayer plates (tip and plug combinations) producing large spray angles are preferable to those which use small spray angles. This is true in express-type boilers, but is a subject of controversy in the B. and W. 2-inch tube boiler. In the sprayer plate type atomizer, sprayer plates of low ratio (08 is the lowest used in the Navy) give larger angles of spray than do those of large ratio (30 is the largest ever used in the Navy). The ratio referred to is a design figure and represents the ratio of combined area of cross section of grooves leading to whirling chamber to the area of the tip orifice. In the tip and plug atomizer it is best to use the smallest size plug with which the desired burner capacity can be attained. In considering this point it must be noted that for a given sprayer plate ratio (or plug for Bureau atomizer) the spray angle increases with increase in diameter of the tip orifice. Therefore the sprayer plate (tip and plug) used for full power will have a spray angle greater than any other combination on board. The shape of the burner opening in the boiler must be designed for this spray angle, therefore at burner capacities of less than full power it is desirable to attempt to hold up the spray angle—by the use of low ratio sprayer plates (or small plugs)—in order to obtain best results with mixing of air and oil in the burner opening in the refractory front wall.

Tip Location: The location of the atomizer tip in the register has an important bearing on mixing of air and oil spray. The tip should be located in the position given in the drawing of the burner assembly furnished the ship. In the B. and W. types of burners it is also important that the tip of the atomizer be located correctly with respect to the impeller. This is ¼ inch on fire room side of the apex of the conical impeller. In this type of burner, the im-

peller and burner are adjustable together along the axis of the register. The impeller is really a functional part of the register. It is important to keep the vanes of the impeller adjusted very closely to that opening given on the drawing of the burner. Withdrawal of the atomizer tip toward the fire room end of the register improves the mixing of the air and oil. If carried too far it will cause the burner to pant.

AIR SUPPLY.—*Panting:* Panting or vibration of the burners is a peculiar condition in oil combustion. While it is due to instability of relation between air and oil velocities to flame propagation, still the basic causes have not been definitely determined. It is known that it can be stopped by increasing excess air, by partially closing the register inlet vanes and sometimes by moving the atomizer toward the furnace (particularly in the B. and W. types of burners). While it is objectionable because of the danger of flare back of flame into the fire room and also the possibility of loosening the anchoring of the refractory, still it is also objectionable from a point of economy because of the fact that it can be, and usually is, stopped by increasing the fire room air pressure. Unless the boiler is making smoke, it should be stopped by using one of the other remedies. Also one should assure that the oil temperature is not too high, as discussed above.

Register Adjustment: Individual adjustment of the vanes in the burner registers will produce lowered efficiency. The vanes of all registers on all boilers in a fire room should be set at the same opening. If it appears that certain burners require special setting of registers different from others in the same boiler room, there is something wrong with the atomization of the burners requiring special register vane settings. It may be that the wrong sprayer plate (tip and plug) is installed, that the atomization is coarse due to wear of the sprayer plate (tip and plug), that the atomizer tip is incorrectly located or not properly centered in the register.

For the conical types of registers, the air inlet vanes should be operated only wide open. In the B. & W. types, an intermediate setting will be required but it should be assured that the opening is not smaller than that which will give good results. The fault in throttling the register inlet vanes is to abnormally increase fire room pressure with resultant loss in boiler room economy due to

increase in blower steam consumption. With these points in mind the operator can ascertain the proper register vane setting. It is best to use the same vane opening for all rates of combustion. A variety of settings leads to confusion.

Blower Control: The forced draft blowers must of course be operated to furnish sufficient air to keep from making objectionable smoke. However, there is still a good deal in their method of operation which seriously affects overall fire room efficiency. The loss next in magnitude to that caused by the use of air for combustion in excess of theoretical requirements, is wasteful blower operation. The two points requiring most attention are as follows: First, hand controlled steam nozzels must be cut in, one after the other, only as necessary to furnish the steam necessary to drive the turbines at the required speed. This involves maintaining a high steam chest pressure whenever a hand nozzle is opened. Each hand nozzle, when opened, should be opened wide and never throttled. For port use, where the blower speed requirements are below that which can be efficiently served by the open steam nozzles under control of the main throttle valve, a separately controlled very small "port" nozzle has been installed on destroyers. Second, when operating more than one blower at a time on a single fire room compartment, it is of great importance that all blowers operating in parallel be maintained very nearly at the same speed. This is important at all ship speeds, but particularly so at full power, where failure to attend to this point may be the cause of failure to obtain the air necessary for full power combustion.

FEEDING BOILERS.

Feeding affects boiler room economy not a little. Maintenance of the maximum possible feed temeprature is of course of primary importance. However, too little attention is ordinarily given to steadiness of rate of feed and position of water level in the steam drum. Unsteady feed rate seriously upsets the operation of the boiler room, and in fact if it be aggravated will affect the entire machinery plant. The more boilers in use the less will be the effect, but it is never inconsequential. Changes in feed rate on change of speed are unavoidable, but are inexcusable at steady speed. For a steady rate of combustion, a change in feed rate

affects the rate of evaporation by changing the incoming feed temperature and the volume of water in the boiler. It will cause fluctuations in steam pressure and a consequent necessary change, even though temporary, in the rate of combustion. The net result is ragged operation with a very detrimental effect on economy due to the impossibility of steadying down and making close adjustment of blowers and pumps. To obtain steady feed it is first essential to set the feed pressure at a definite amount in excess of the main steam pressure. In port, a lower excess pressure can be used than when underway. But the excess pressure to be carried should be set by the chief engineer and not left to the discretion of the feed pump tender. Note that the actual pressure on the feed line will vary with the choice of current operating boiler pressure, if that be below safety valve setting. Excess pressures suggested are,

Full power and maneuvering, 75 pounds.

Cruising speed, 50 pounds.

Port, 30 pounds.

A governor to maintain a definite pressure in excess of boiler pressure is a real asset in attempting to attain maximum fireroom efficiency.

The position of the water level maintained in the steam drum of water tube boiler is of considerable importance not only to safe operation, but to efficiency also. The higher the water level the more safety is attained. The lower the level is carried, but always above the bottom nut of the gauge glass, the dryer will be the steam produced. This latter becomes important only when the boiler is operated at above about three-quarters of full power. It is most important on boilers fitted with superheaters as the superheat will vary with the quality of the steam produced. Under ordinary circumstances the water level should be carried, as shown by the gauge glass, at the horizontal diameter of the steam drum. The water level to be carried should be prescribed by the Chief Engineer and not left to the whim of the water tender on the watch. A good water tender should be capable of maintaining the water level within plus or minus one inch of that desired, except temporarily when marked changes are made in ship speed. Even then the variation should not exceed plus or minus five inches. Automatic feed water regulators are much needed in the Naval Service.

STEAM PRESSURE.

The steam pressure to be carried on the boilers should be regulated to suit the requirements of the main propelling engines. The lower the steam pressure the less will be the losses in smoke pipe gases and in boiler and steam pipe radiation. The designed steam pressure on any ship is set by full power requirements. At ship speeds less than full power, and in port, the pressure necessary on the main and auxiliary lines is only that required for proper setting of steam chest pressure on the machinery in use. Destroyers, and even battleships, underway at cruising speeds, operate more efficiently at a boiler pressure somewhat less than safety valve setting. In port, the economy of lowered pressure is quite marked. The steam pressure to be carried from hour to hour should be set by instructions from the Chief Engineer, to suit most economically the operating conditions obtaining.

APPROXIMATE HEAT AND WATER LOSSES IN MACHINERY SPACES.

By J. K. Esler, Lieut. Comdr., U. S. Navy, Member.

During the trials of a light cruiser the temperature rise from 30 degrees F. outside air to over 100 degrees F. in the engine rooms was considered excessive. It appeared probable that the machinery spaces would be untenable at high powers in the tropics. In view of this and the comparatively satisfactory performance of other vessels of the class under similar conditions an attempt was made to determine the cause of excessive rise in temperature and the approximate amount of heat and water losses.

Engine rooms showed some steam leakage from glands but this was not excessive in comparison with conditions observed on other vessels in service. Clothing and lagging on steam pipes, valves, turbines, etc., was practically complete and of the thickness and materials required by specifications. Some few valve bonnets, flanges, steam cylinder heads of pumps and surfaces not required to be lagged by the contact remained uncovered.

Wet and dry bulb psychrometers were improvised and located in various parts of the forward engine room. Readings were taken at half-hour intervals during the four-hour full power trial. Considerable difficulty was experienced in getting consistent readings due to drafts from blower inlets and proximity of radiating surfaces, and finally the readings of a psychrometer located above the working platform, clear of direct contact with incoming air, reasonably distant from bulkheads and hot surfaces, and yet where air was free to circulate, was taken as most indicative of the general conditions prevailing in the engine room.

A method of computing heat and water losses was developed later, and as it is one which may be followed by engineer officers

on other ships it is given in detail for their information. The method involves determination of the heat and moisture contained per pound or per cubic foot of ventilating air under conditions obtaining on deck and in the engine room. The difference represents the heat and moisture taken up by the air in the engine room due to radiation losses and steam leakage. If the capacity of engine room ventilating sets can be determined the amount of such losses may be computed. But even without this the results are of value for comparison with tests made in the same engine room at other times and for comparison of results obtained in different engine rooms or on similar installations in ships of the same class.

The humidity and psychrometer chart by Carrier, on page 342 of Mark's Mechanical Engineers' Handbook, was enlarged and used for determination of humidity and B. T. U. content of the air. Conditions obtaining during the full power trial were as follows:

Outside air—dry bulb, 30 degrees; relative humidity, 90 per cent.

Engine room—dry bulb, 104 degrees; wet bulb, 90 degrees; relative humidity, 63.5 per cent.

B. T. U. per pound air outside conditions	10.8
B. T. U. per pound air in engine room	57.0
B. T. U. gained per pound of air	46.2

Assuming baromter at 30 (barometer readings were not taken at the time of the tests), the weight of air per cubic foot was computed from the formula on page 339, Mark's Handbook. The pounds per cubic foot of outside air $= 1.326\ (B/T) - 0.501\ (he/T) = .08124 - .00017 = .08107$.

Ventilating blowers were rated at 20,000 cubic feet per minute and were operating at full capacity.

Weight of air delivered to one engine room per minute $= 20{,}000 \times .08107 = 1621$ pounds.

B. T. U. absorbed by ventilating air in one engine room per minute = 1621 × 46.2 = 74,890 B. T. U. per minute.

This is equivalent to a heat loss of 1766 horsepower, found by dividing the B. T. U. per minute by 42.4.

For determination of moisture taken up by the ventilating air in one engine room under full power conditions, table 40, page 340, of Mark's Handbook was used—

Water vapor per cubic foot of saturated air at 30 degrees = 1.973 grains.

Water vapor per cubic foot of air at 90 per cent relative humidity = 1.7757 grains.

Water vapor per cubic foot of saturated air at 104 degrees = 22.35 grains.

Water vapor per cubic foot of air at 63.5 per cent relative humidity = 14.192 grains.

Increase in water vapor per cubic foot ventilating air = 12.416 grains.

Moisture taken up by ventilating air in one engine room =

$$\frac{20{,}000 \times 12.416 \times 60}{7000 \times 8.3} = 256 \text{ gallons per hour}$$

In order to check this data at other speeds than at full power, psychrometer readings were taken during standardization runs at the following speeds: 20 knots, 28½ knots and 32 knots. Data obtained was as follows:

20 knots—85 degrees wet bulb—100 degrees dry bulb—55 per cent relative humidity.

28½ knots—88 degrees wet bulb—102 degrees dry bulb—57.5 per cent relative humidity.

32 knots—91 degrees wet bulb—105 degrees dry bulb—59 per cent relative humidity.

Outside air—29 degrees wet bulb—33 degrees dry bulb—60 per cent relative humidity.

Computations were made as for the full power run and are tabulated for comparison:

	20 Knots	28¼ Knots	32 Knots	F. P.
Temperature rise in engine room	67 degrees	69 degrees	72 degrees	74 degrees
Relative humidity in engine room	55 per cent	57½ per cent	59 per cent	63.5 per cent
B. T. U. gained per pound of air	37	40.45	44.2	46.2
B. T. U. absorbed by air in 1 engine room per minute	59,977	65,659	71,648	74,890
Equivalent horse-power loss	1415	1548	1690	1766
Grains moisture gained per cubic foot air	9.644	10.816	12.236	12.416
Gallons water lost to air per hour	199.2	223.4	252.7	256

For further comparison data was taken during the full power trials of another vessel of the same class, but built by a different contractor and equipped with double standard thickness of lagging on main steam piping. This vessel was operating at approximately 100 per cent efficiency based on engineering competition allowances and was considered to be in excellent condition as regards radiation losses and steam leakage. Data was taken by sling psychrometer in both engine rooms at hourly periods throughout the trials. The full power speed corresponds closely to the 32-knot run of the other vessel due to the limitation placed on speed.

Temperatures were as follows:

	Outside air	Forward engine room	Aft. engine room
Dry bulb	66 degrees	105 degrees	98 degrees
Wet bulb	62 degrees	86 degrees	81 degrees
Relative humidity	80 per cent	47 per cent	49 per cent
Barometer	30.28	.....	.....

Ventilating blowers were operated at full capacity and it is assumed that they were delivering 20,000 cubic feet of air per minute to each engine room.

Air under the conditions obtaining on deck weighed .0759 pound per cubic foot (from the formula on page 339 of Mark's Handbook: D—1.326 (B/T)—0.501 (he/T) using table 40:

$$= 30.28 \times .002523 = .0764$$
$$(-) \quad .80 \times .000614 = .0005$$
$$.0759 \text{ lbs.}$$

$20{,}000 \times .0759 = 1518$ pounds of air per engine room per minute. Tabulating the results:

	Forward engine room	After engine room
Temperature rise in engine room	39 degrees	32 degrees
Relative humidity	47 per cent	49 per cent
B. T. U. per pound of air in engine room	48.85	43.65
B. T. U. per pound air outside = 27.5	.....	.....
B. T. U. gained per pound air	21.35	16.15
B. T. U. absorbed by air per minute	32,409	24,515
Equivalent horsepower loss	764.3	578.0

Water vapor per cubic foot saturated air at 66 degrees = 7.048 grains.

Water vapor per cubic foot at 80 per cent relative humidity = 5.638 grains.

Water vapor per cubic foot in engine room air: Forward engine room = 10.805 grains; after engine room = 9.207 grains.

Water vapor gained per cubic foot air: Forward engine room = 5.167 grains; after engine room = 3.569 grains.

Water absorbed by air per hour—Forward, 106.7 gallons; after, 73.71 gallons.

To summarize the results of the foregoing:

	First light cruiser				Second light cruiser	
	20 Kts.	28.5 Kts.	32 Kts.	F. P,	Full power run	
					Fwd.	Aft.
Temperature rise degrees	67	69	72	74	39	32
H. P. Loss	1415	1548	1690	1766	764	578
Water Loss (gallons per hour)	199.2	223.4	252.7	256	106.7	73.7

It is generally conceded among operating engineers that it is more economical to operate at low steam pressure, retaining only sufficient difference between boiler pressure and steam chest pressure to allow for emergencies and expected changes in speed, than to operate at high steam pressures and gain the advantage of drying the steam by wire drawing through the throttles. This condition has generally been ascribed to radiation losses, without attempting to evaluate the losses quantitatively. It is well known that make up feed should be reduced as much as possible by the elimination of all steam leaks, but few if any attempts have been made to measure the losses beyond recording amount of make up feed used. In the tests conducted on the full power trials of the second ship there was nothing except a slight difference in temperature to indicate that the after engine room was more efficient than the forward; yet the results show the forward engine room to have had 13.2 per cent more heat losses and 14.5 per cent more steam leakage than the after engine room.

The necessity for adequate lagging and careful elimination of steam leaks is shown in the comparative performance of the two ships. In making these computations it was expected that a considerable difference would be shown, but the results exceed all expectations. There are ships in service which are much less adequately lagged and have more visible steam leaks than noted on either of the ships observed.

It is believed that the next step toward improved economy in the present type of engineering plant must be toward reduction of the so-called radiation losses and the elimination, in so far as possible, of steam leakage. In this connection it is believed that

much can be accomplished when the machinery is installed. For instance: Most steam piping is secured by metal straps which make a metallic connection between the steam pipe and the framing of the ship. This framing is practically integral with bulkheads or decks, which form ideal surfaces from which radiation may take place. The straps or brackets form very good pathways along which heat may flow by conduction from the hot pipe to the radiating bulkhead.

To find out to what extent this particular feature was true, two thermometers were secured to the straps supporting a main steam line where it passed through a dead fireroom. Fireroom temperature in the open at the height of the steam pipe was 98 degrees. Temperature of steam in the pipe was 394 degrees. The thermometer fixed to the strap in the open fireroom where there was a free circulation of air registered 132 degrees. The other thermometer located on a strap where air circulation was restricted registered 162 degrees. Steam pipe was lagged with double the standard thickness of lagging and thermometers were over two feet from the pipe.

This is merely one suggestion. It is quite possible that operating engineers who make a study of the various forms of radiation losses may discover other pathways of heat loss quite as important as this. It seems reasonable that whenever possible the supporting straps and brackets should be heat insulated either from the steam pipe or from the framing of the ship. This procedure is already followed in securing refrigerating piping. There may be some material suitable for use in insulating turbines, boilers, evaporators, etc., from the hull of the ship.

Radiation from steam cylinder heads, valves, flanges, etc., which are not covered is one of the chief sources of radiation loss. The rate of radiation depends on the color and smoothness of the surface and may be as great as the fourth power of the difference in temperature between the radiating surface and surrounding objects. If these surfaces were merely covered with a thin asbestos sheet in order to retain a film of air next to the surface, the radiation would be very much reduced. The more adequate the covering the more radiation losses will be reduced. Dark colored

surfaces radiate more than light colored surfaces, and dull surfaces more than polished surfaces.

Losses due to radiation and which can be determined at least approximately by the method shown, consist of a constant loss, apparent at all speeds, and an additional loss which increases with the power being developed. In the tests and computations made the heat taken up by ventilating air in one engine room at the various speeds, converted to equivalent horsepower, is in proportion to the power developed as follows (figures are approximate):

	20 Kts.	28½ Kts.	32 Kts.	F. P.
Power developed in one engine room	6000	20,300	34,400	48,000
Horsepower loss to air	1415	1548	1690	1766
Per cent of horsepower loss to horsepower developed	23.6	7.6	4.9	3.7

The fact that the percentage increases at lower powers indicates the constant loss. It is believed that at still lower powers a much greater percentage of power developed is lost. This is particularly important at economical speeds and could be much reduced by more efficient means to prevent radiation losses.

Much research work has been done to determine the best materials to use for pipe covering and the proper thickness to use for different conditions. The best material appears to be that which holds the most air. The thickness depends on temperatures, as it may be quite possible to get a covering so thick that heat is wasted in heating it, or so thin that heat passes through with little restriction. Lagging should be sufficiently tight so that there is no current of air through it. Air is one of the best non-conductors of heat known. It may be desirable to secure the pipe covering in such a way as to leave an annular air space between pipe and cover. Economy would be gained not only by reduction of radiation losses, but also by reducing the power required for ventilating. There is room for considerable more research work. It is possible that it may prove of value to enclose main steam piping, and some of the other surfaces which are at present covered, with air-tight

jackets, and to maintain a partial vacuum around these heated surfaces to reduce radiation losses. As progress points toward higher steam pressures and temperatures, the problem becomes more important. Present types of lagging are inadequate to such an extent that the pressures available cannot be used to full advantage when operating at low powers.

If the same loss existed in both engine rooms of the first ship on which this test was conducted, the heat loss in the two engine rooms, as converted to horsepower, would be sufficient to drive the ship at the following speeds:

Loss at 20 knots = 2830 H.P. = S.H.P. used at 12.66 knots.
Loss at 28½ knots = 3096 H.P. = S.H.P. used at 12.77 knots.
Loss at 32 knots = 3380 H.P. = S.H.P. used at 13.22 knots.
Loss at full power = 3532 H.P. = S.H.P. used at 13.66 knots.

A NEW THRUST MEASURING DEVICE.

By H. T. Herr,* Member.

Shaft dynamometers of several types have been in use for many years on large vessels, both Naval and merchant. Owing to the delicate nature of these instruments and the difficulty of their calibration, they have never been entirely satisfactory.

In the early development of turbine reduction gearing for marine propulsion in the United States, initiated by the late George Westinghouse, a feature of one of the designs devised by him was a dynamometer incorporated in the construction of the frame carrying the pinion of the reduction gear. By supporting the pinion frame on hydraulic pistons and measuring the oil pressure required so to support it, quick and accurate determination of the power transmitted was made possible, as well as automatic alignment of the pinion to the gear. For commercial reasons, irrelevant to this discussion, the manufacture of this device was practically discontinued, although quite recently it has been employed for special purposes in several instances, and a number of turbo reduction gears were fitted and put into service a number of years ago embodying this construction with entirely satisfactory operating results.

The demand for accurate and reliable means of checking ship performance has always existed and has become more insistent as refinements in engineering have developed in the design and construction of ships of high speeds and powers.

In lieu of the costly and delicate shaft torsion dynamometer, simpler propeller shaft thrust measuring devices have been developed, which, although they do not give the same information as the torsion dynamometer, furnish valuable and important data concerning the condition of the hull and a more or less direct means of judging the efficiency of hull and propeller design.

* Vice-President, Westinghouse Electric and Manufacturing Company.

An instrument of this nature is shown in the accompanying illustration. It is incorporated in the structure of the Kingsbury propeller thrust bearing housing, and comprises essentially: a source of oil pressure, automatically maintained, to balance the propeller thrust against pistons forming the Kingsbury thrust bearing shoe supports; accurate and dependable gauges for noting or recording the oil pressure required; and suitable valves and linkage to automatically control the admission of oil under pressure to the thrust bearing casing in ahead or astern operation of the ship for the purpose of exactly balancing the propeller thrust.

In the illustration, "A" represents the outer casing and "B" the inner housing containing a Kingsbury thrust bearing. The thrust bearing segments are indicated at "C." "D" is the thrust collar. The housing "B" is free to move endwise within the casing "A" a small distance. It will be noted that the upper part of the housing "B" carries a projection "E" which engages multiplying levers "F" and "F_1" (the latter shown in the insert) connected through stems "G" and "G_1" to valves "H" and "H_1," respectively. Obviously the extent of movement of the valves in response to a given displacement of "B" is in proportion to the lever ratio of "F" and "F_1." The valves are made sufficiently large in area that only a few thousandths of an inch of end motion is required to admit full oil pressure to the casing. The valve chambers are connected by pipes to the spaces at the ends of the casing "A." These spaces are normally filled with oil at the bearing supply pressure of about five pounds gauge.

Under normal conditions with the vessel under way, the housing "B" will be held against the forward end of the casing (the left-hand end in the illustration) by the action of the propeller thrust. In this condition, the valve "H" would be open and the valve "H_1" closed. When it is desired to measure the thrust, it is only necessary to start a small high pressure oil pump, provided for the purpose, which supplies oil under pressure to the valve chamber. Valve "H" being normally open, as described above, will admit high pressure oil to the forward face of housing "B," which will be moved aft as the oil pressure overcomes the thrust. Obviously, however, motion of "B" in this direction will cause a corresponding closing movement of valve "H," throttling the supply of oil

TO PROPELLER SHAFT
TO REDUCTION GEAR

to the casing. Hence a condition of equilibrium will exist when the housing "B" just floats in casing "A," supported by the oil pressure, which in this condition must exactly balance the propeller thrust. An accurate reading of the oil pressure gauge and the application of a constant will then give the total propeller thrust at that moment being transmitted to the ship's structure through the thrust bearing.

If the ship were moving astern, the action of the device would be reversed, the oil passing through valve "H_1" to the aft end of casing "A," and the oil gauge reading would then register the astern thrust.

The value of this device can be enhanced by employing a recording pressure gauge, in addition to the indicating gauge. Continuous records of the propeller thrust can then be obtained from which can be deduced much valuable information concerning the rate of fouling of the ship's bottom, her behavior under varying sea conditions, etc.

For a complete study of performance of a vessel powered with direct-connected reciprocating steam engines, steam turbines, or Diesel engines, obviously the ideal combination would be a thrust measuring device and a torsion meter or dynamometer. For geared turbine or geared Diesel vessels, the thrust measuring device and the pinion frame dynamometer offer the simplest and most reliable combination; while in electric-drive vessels, the thrust meter and the electrical instruments for measuring the power input to the propeller shaft make available complete data on the condition of both machinery and hull.

It is interesting to note in passing that the author devised this thrust measuring device some fifteen years ago for the express purpose of getting direct and accurate knowledge of the propulsive efficiency of ships in combination with the Westinghouse gear and dynamometer hydraulic floating frame arrangement for turbine reduction gear propulsion, but that neither of these devices has until recently been sufficiently appreciated to merit an installation on board ship for the purpose of actually studying first-hand the characteristics of hull and propeller design and construction for the best propulsive efficiency. The general tendency heretofore

has been to rely on model tank experiments for the hull and torsion dynamometers for the machinery. It is believed that the initial installation of thrust dynamometer as above described and as manufactured by the Westinghouse Electric & Manufacturing Company, undertaken by the Shipping Board, will yield important data on ship propulsion coefficients.

DEVELOPMENT OF B. & W. BOILERS FOR NAVAL VESSELS.

TESTS OF FOUR TYPE BOILERS AT THE FUEL OIL TESTING PLANT.

By R. C. Brierly,* Associate Member.

A large number of boilers have been tested at the Fuel Oil Testing Plant since its inception; among these have been four designed and manufactured by the Babcock & Wilcox Co. Each of these four have been subjected to lengthy tests, the first two being fired during separate tests by many different kinds of burners. At least one series of official runs has been conducted, however, on each of the four using Babcock & Wilcox burners. Official runs on the first of these boilers discussed were run with Babcock & Wilcox *Pennsylvania* type burners—a forerunner of the present Cuyama;† the last three with Babcock & Wilcox Cuyama burners. The results obtained on these boilers, the last three fired by substantially the same, though somewhat modified burners, furnish a highly interesting comparison.

FIRST BABCOCK & WILCOX MARINE TYPE OIL BURNING BOILER INSTALLED AT THE FUEL OIL TESTING PLANT.

The first of the four boilers to be considered was installed at the Plant in the year 1912, in order that the efficiencies and operating characteristics of this type of boiler, when fired with various fuel oil burners and equipped with previously tested auxiliaries, could be accurately ascertained.

* Lieutenant-Commander (E-V) U. S. N. R. Head Test Engineer, Fuel Oil Testing Plant.

† Note: Development of the Cuyama burner was begun prior to the year 1915 by the Babcock & Wilcox Co. for use under forced and natural draft conditions at rates of firing not in excess of 1000 pounds of fuel oil per hour. It was originally known as the Peabody burner, having been designed by Mr. E. H. Peabody, but since 1919 has been called the Cuyama burner—as such it is not to be confused with the Peabody burners now manufactured by the Peabody Engineering Corporation, 110 E. 42nd St., New York, N. Y. Since the inception of the Cuyama burner it has been somewhat modified, its manufacture, and operation simplified and its efficiency increased.

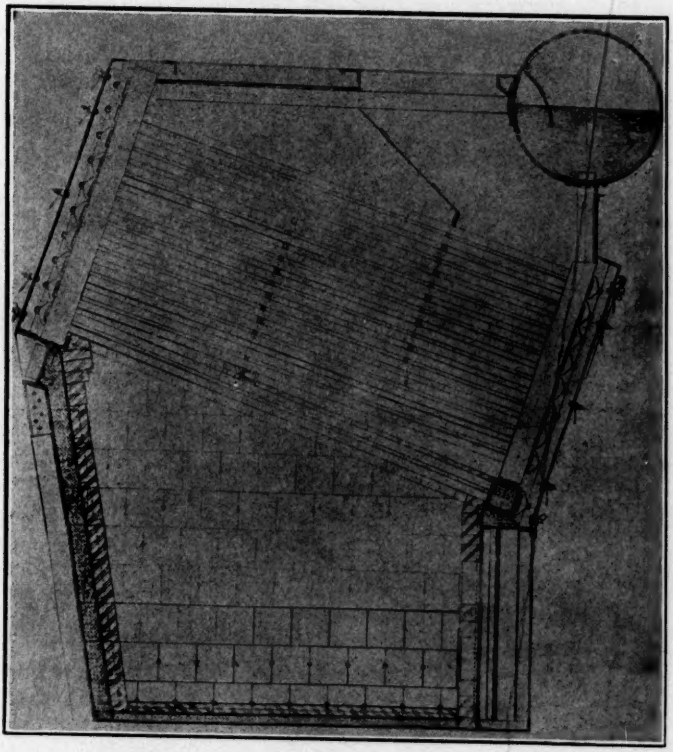


FIGURE 1.

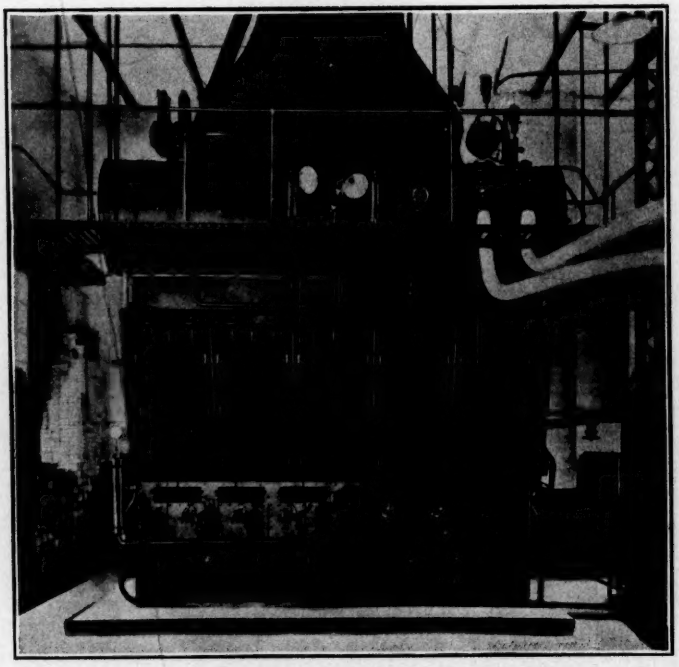


FIGURE 2.

BABCOCK & WILCOX BOILER, U. S. NAVAL OIL-FUEL TESTING-PLANT, NAVY YARD, PHILADELPHIA, PA.

Although a considerable number of coal fired boilers of the same general type had been in Naval use for some time, the projected installation of this boiler, oil fired, aboard first line battleships, numerous smaller fighting craft, and auxiliaries, rendered determination of complete and accurate oil fired data imperative. As the first marine installation of this type and size of boiler was made in the battleship *Oklahoma,* the boiler came to be known as the Babcock & Wilcox *Oklahoma* type. A cross section of this well known steam generator is shown in Figure 1, particulars of the unit as installed at the Fuel Oil Testing Plant are as follows:

Duplicate of boilers installed in U. S. S. *Oklahoma*	No. 295 W. P.
Height—top of foundation to drum over rivets, feet and inches	14–0
Width—over drum manhole fittings, feet and inches	15–4
Length—face of drum feed nozzles to extreme rear, feet and inches	12–10
Space occupied, cubic feet	1580
Distance front plate to bulkhead required for tube renewal, feet and inches	9–7½
Heating surface 1 boiler, square feet	4000
Weight 1 boiler dry, pounds	69,100
Weight 1 boiler wet (steaming level and temp.), pounds	82,650
Number of sections	22
Number of groups 4–2 inch diameter tubes in each header	9
Total number of 2 inch tubes	792
Length of tubes between headers, feet and inches	9–0
Inside diameter of the steam drum, feet and inches	3–0
Furnace volume, cubic feet	445
Superheater heating surface	0

A view of the boiler showing the installation of thirteen burners —then known as *Pennsylvanias*—on the boiler front plate but with a modified double front is shown in Figure 2.

A series of runs was conducted with the boiler and burner installation shown. The principal results obtained during official runs are shown in Figure 3.

The range of rates officially run during this test was from approximately 0.225 pound fuel oil per square foot of B. W. H. S. per hour (900 pounds of oil per hour on this boiler) to 1.23 pounds fuel per square foot of B. W. H. S. per hour—4880 pounds oil per hour.

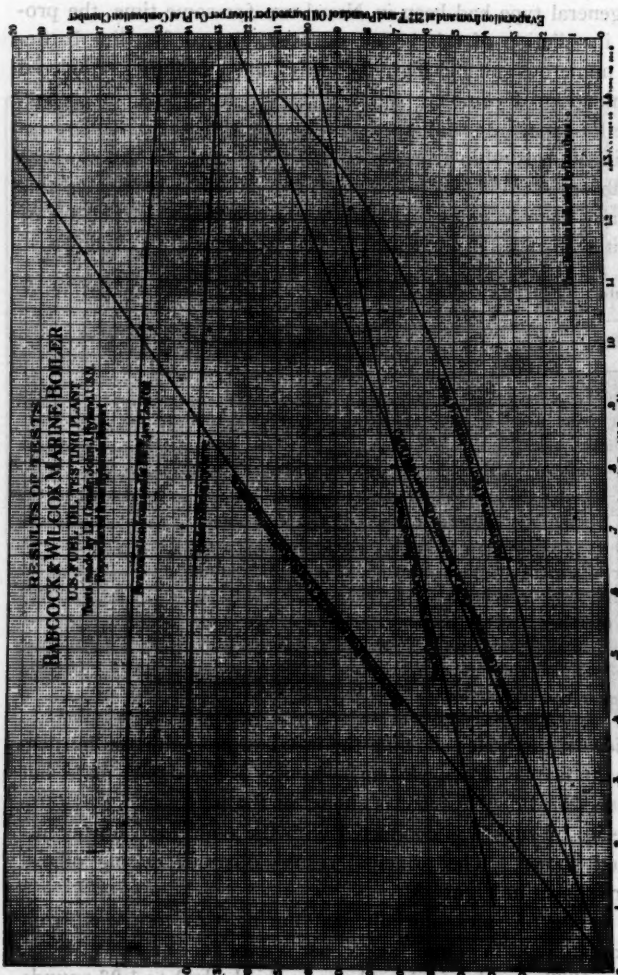

BABCOCK & WILCOX MARINE BOILER
Pounds of Oil Burned per sq. Ft. H.S. per hour
Boiler Efficiency per Cent
Steam Temperature, Degrees F.

FIGURE 3.

The overall (boiler, burners, and furnace) efficiency ranged from a maximum of approximately 80 per cent at the lowest rate to a minimum of 76.5 per cent at the highest rate. The equivalent evaporation was about 16 pounds per pound of oil—14,400 pounds total—at the 0.225 pound rate and 15.3 pounds per pound of oil—74,664 pounds total—at the 1.23 pounds combustion rate. It is to be noted that the temperature of gases leaving the boiler ranged from approximately 440 degrees F. at the lowest rate to 850 degrees F. at the highest rate. The fireroom air pressures necessary with the combinations of burners used ranged from 1.45 inches (of water) to 7.90 inches.

The Cuyama burner, substantially as used on runs of the last three boilers (except for impeller changes and alterations in minor details) is shown in Figure 4.

The results were highly gratifying at the time of this test and they were the subject of lengthy discussion and comment. A sentence taken from an article contemporary with this test read as follows: "This boiler holds the world's record for economy and capacity, having evaporated 18.7 pounds of water per square foot of heating surface per hour and 15.3 pounds of water per pound of oil (both F. & A. 212 degrees F.) when burning 1.23 pounds of oil per square foot of heating surface per hour.

This type of boiler had—and has—very widespread use both in Naval vessels and those of the merchant service. The boiler upon which these tests were run is still in service at the Fuel Oil Testing Plant, supplying steam for auxiliary purposes and many relatively minor tests of pumps, regulators, etc.

SECOND BOILER MANUFACTURED BY THE BABCOCK & WILCOX CO. INSTALLED AT THE FUEL OIL TESTING PLANT.

The second boiler of that company to be submitted for test at this Plant was built and installed five years later in 1917. The principal purpose of testing this boiler was to determine its merits as a steam generator for battleships and Scout Cruisers. Many other purposes—such as the utility of downcomers and comparative efficiencies possible with various types burners, to mention only one or two—were also served by data from operation of this boiler.

This boiler was a modified White-Forster Express or A type boiler (now known as the Babcock & Wilcox Express type) with superheater units in the center of each bank of tubes. The appearance of the boiler as originally installed at the Plant, with eleven Cuyama burners on the boiler front plate is shown in Figure 5. Front elevation of the boiler, partly in section, with Mayflower-Lodi burners—afterwards tested at the Plant—on the boiler front, is shown in Figure 6.

The principal particulars of this boiler were as follows:

Working pressure, pounds	280
Height—bottom brick pan to top of drum over rivets, feet and inches	15–9–9/16
Width—over side casings, feet and inches	15–2–1/4
Length—over drum manhole fittings, feet and inches	15–4–1/4
Space occupied, cubic feet	2775
Distance front plate to bulkhead required for superheater tube renewal, feet and inches	22–0
Heating surface one (1) boiler, square feet	7565
Superheating surface one (1) boiler, square feet	753
Weight boiler and superheater dry, pounds	94,650
Weight boiler and superheater wet (steaming level and temperature), pounds	109,800
Tubes all 1 inch O. D. 4032 in number, 109 mils thick.	
Superheater 2 units 64–1 inch O. D. return tubes per unit.	
Superheat, degrees F.	50–70
Steam drum—54 inches I. D. 14 feet 6 inches long outside.	
Water drums (2) 24 inches I. D. 13 1/4 inches long outside.	
8.4 inches downcomer tubes, 4 to each mud drum, at front of boiler between inner and outer casing.	

During the year 1918 a series of more than 100 runs was made on this boiler with the eleven Cuyama burners as installed, the results of these runs surpassed any that had been previously obtained at the same rates on boilers tested at the Plant. In the year 1927, however, after certain advantageous modifications in atomizers and impellers had been made by the Babcock & Wilcox Co. and the technique of burner arrangement, sprayer plate selection and operation had been considerably improved by the Fuel Oil Testing Plant, a series of runs was made with nine modified Cuyama burners. This series showed considerably higher efficiencies than those of 1918 in spite of the fact that the boiler was now ten

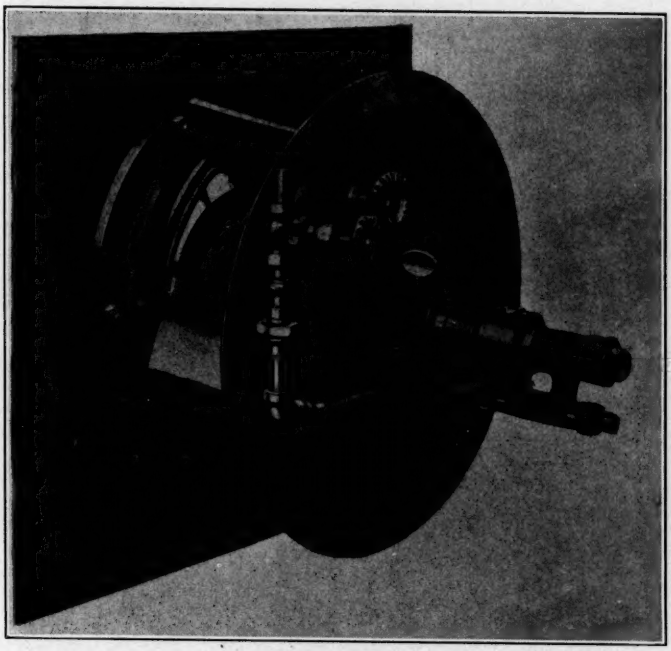


FIGURE 4.

CUYAMA TYPE OF BABCOCK & WILCOX MECHANICAL ATOMIZING OIL BURNER.

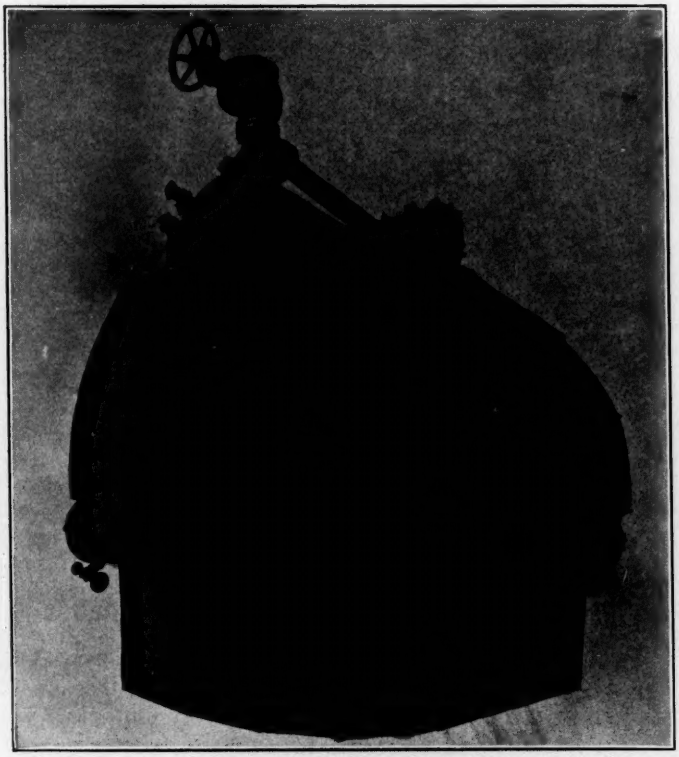


FIGURE 5.

BABCOCK & WILCOX EXPRESS TYPE BOILER. HEATING SURFACE, 7565 SQUARE FEET; SUPERHEATING SURFACE, 753 SQUARE FEET; WORKING PRESSURE, 280 POUNDS. INSTALLED AT THE FUEL OIL TESTING PLANT, U. S. NAVY YARD, PHILADELPHIA, PA.

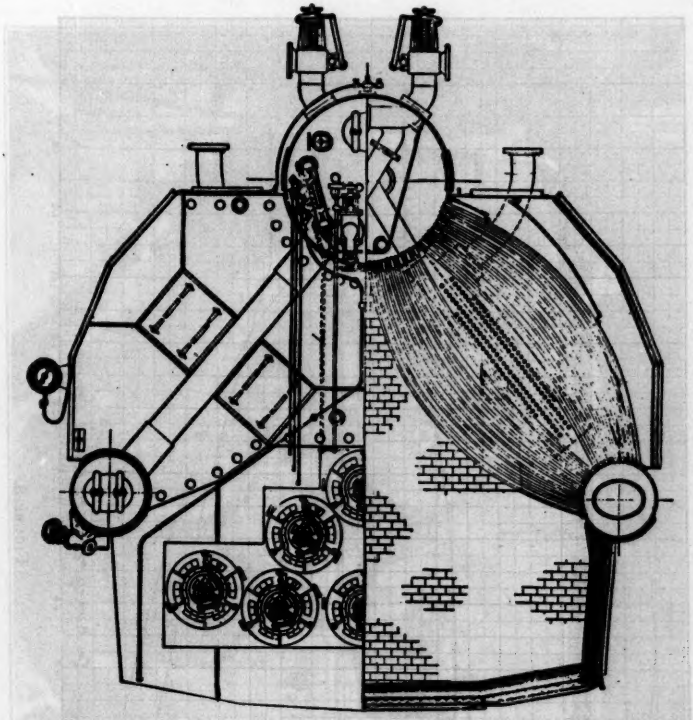


FIGURE 6.

years old and had been in continual use for test purposes—at high, as well as ordinary, rates of combustion—during that period.

The appearance of the boiler front with the nine Cuyama burner installation used during this second Cuyama series is shown in Figure 7. Curves showing principal results obtained during the series of 1927 are shown in Figure 8.

The range of rates run during this series was from 0.103 pound fuel oil per square foot—982 pounds per hour on this boiler—to 1.11 pounds fuel oil per square foot—8406 pounds per hour on this boiler. The best overall efficiency obtained at the 0.103 pound rate was 81.43 per cent—equivalent evaporation was 15,581

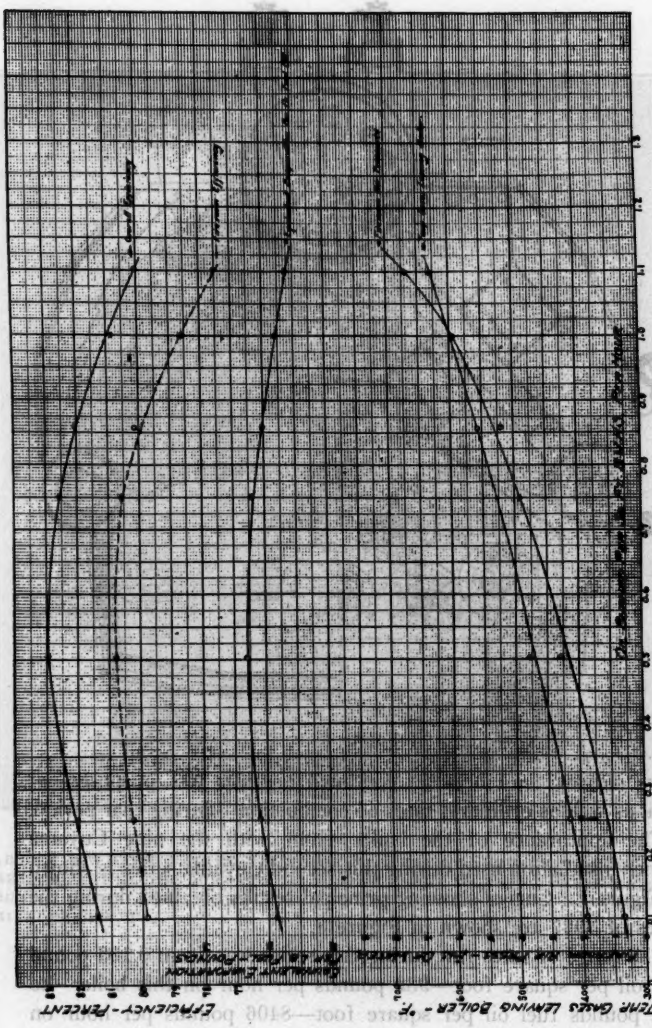


FIGURE 8.

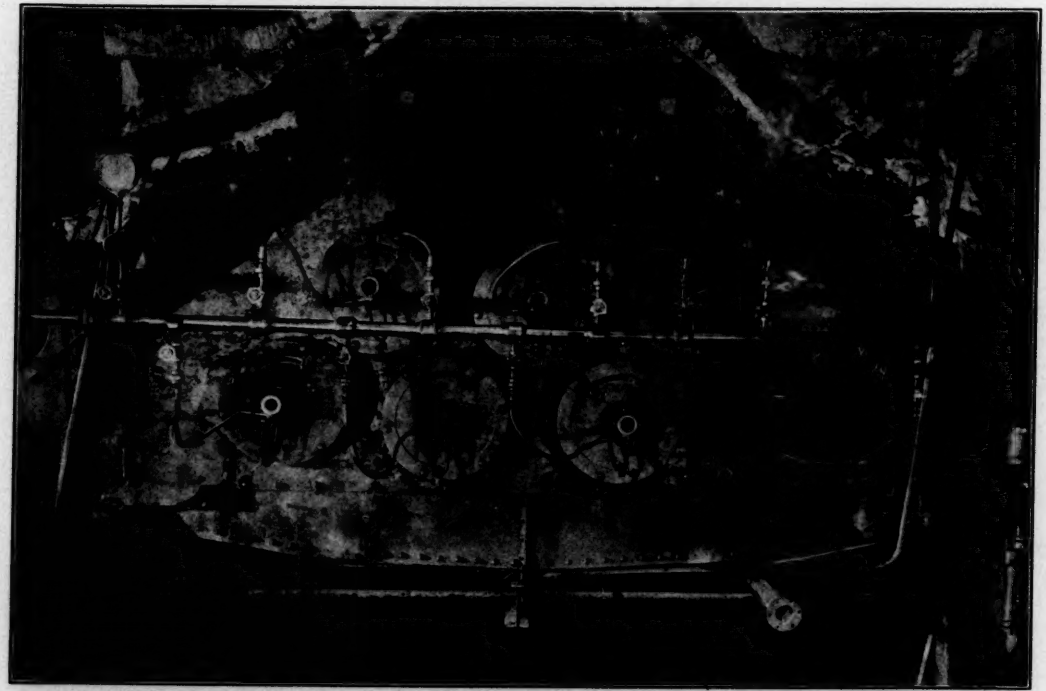


FIGURE 7.

JUNE 3, 1926. INSTALLATION OF CUYAMA BURNER ON WHITE FORSTER BOILER, F. O. T. P., NAVY YARD, PHILADELPHIA, PA.

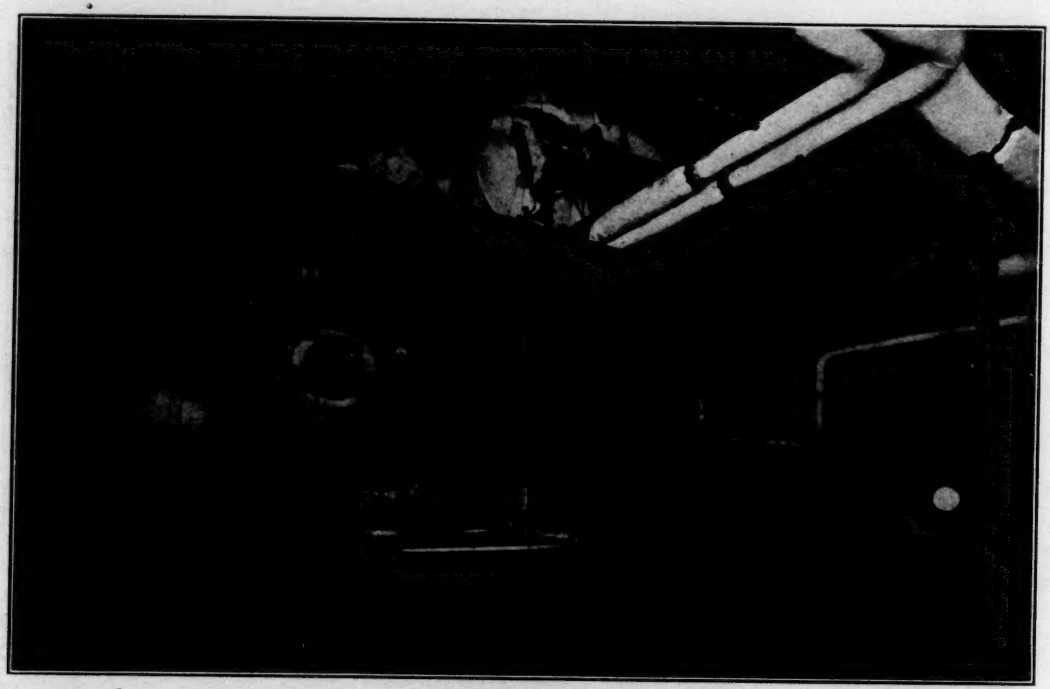


FIGURE 9.
CRUISER BOILER FIREROOM, F. O. T. P.

pounds; at the 0.25 pound rate, 82.0 per cent—equivalent evaporation was 30,571 pounds; at the 0.5 pound rate, 82.92 per cent—equivalent evaporation was 61,947 pounds; at the 0.75 pound rate 82.55 per cent—equivalent evaporation was 91,762 pounds; at the 0.858 pound rate 81.99 per cent—equivalent evaporation was 103,798 pounds; at the 1.0 pound rate 80.85 per cent—equivalent evaporation was 119,392 pounds; at the 1.1 pound rate 80.04 per cent—equivalent evaporation was 131,312 pounds. It is to be noted that the highest efficiency was obtained at the 0.5 pound rate.

These results for express boilers without air heaters or economizers and under closed fireroom forced draft conditions, at the time, again constituted a Plant record.

The fireroom air pressure necessary for operation of this boiler, with the numbers of burners, arrangement of burners and sprayer plates used ranged from approximately 0.7 inch at the 0.103 pound rate to 7.4 inches (of water) at the 1.1 pound rate, temperature of gases leaving the boiler ranged from 395 degrees F. at the 0.103 pound rate to 630 degrees F. at the 1.1 pound rate. Superheat at the 0.103 pound rate was 3.6 degrees F., at the 1.1 pound rate 61.8 degrees F.

The highest combustion rates at which this boiler has been operated have been in the vicinity of 1.5 pounds fuel oil per square foot of B. W. H. S. per hour. This rate has been maintained for two or three hours on more than one occasion. Results obtained from operation at this rate with eleven Cuyamas of the original burner layout were, in brief, as follows:

Combustion rate pounds per square feet B. W. H. S. per hour	1.501
Oil per hour total	11,355
Oil per burner, pounds	1032
Superheat, degrees F.	80.8
Fireroom air pressure (inches of water)	9.5
Total equivalent evaporation	171,971
Equivalent evaporation per pound fuel oil	15.145
Boiler horsepower developed	4985
Per cent rating, per cent	659
Overall (boiler, furnace and burners) efficiency, per cent	76.15
Oil burned per cubic foot, furnace volume per hour	15.120

THIRD BOILER MANUFACTURED BY THE BABCOCK & WILCOX CO. INSTALLED AT THE FUEL OIL TESTING PLANT.

The third boiler of the same company's manufacture to be subjected to test at the Plant was built and installed approximately ten years after the second had been delivered. This boiler, set up in 1927, was the first of a series constructed for the new Scout Cruisers. This boiler is a Babcock & Wilcox Express or A type boiler without superheater units; it is much larger than the previous boiler of the same type tested.

The appearance of this boiler as installed in a new all-steel fireroom at the Plant is shown in Figure 9; front elevation partly in section is shown in Figure 10. Thirteen Cuyama burners were installed on the boiler front plate as shown in the illustrations.

The principal particulars of this boiler are as follows:

Working pressure, pounds	300
Height (bottom brick pan) to top of drum over rivets, feet and inches	16–11–1/2
Width—over side casings, feet and inches	17–0
Length—over drum manhole fittings, feet and inches	17–1–1/2
Space occupied, cubic feet	3927
Distance front plate to bulkhead required for tube renewal—Note all tubes remove through drum so minimum space required for firing will be O. K.	
Heating surface 1 boiler, square feet	11,880
Weight 1 boiler dry, pounds	104,900
Weight 1 boiler wet (steam level and temperature), pounds	123,200
Superheater heating surface	0
Furnace volume cubic feet	1137
Number of rows of tubes per bank	24
O. D. of tubes A. & B. rows both sides—inches	1–1/4
Thickness of 1–1/4 inch tube walls	0.109
O. D. of tubes of all other rows, inches	1
Thickness of 1 inch tube walls	0.095
Number of tubes in A and B rows both sides	332
Total number of tubes in all other rows	4598
Grand total number tubes all rows	4930

During the year 1928 and part of 1929, a series of approximately one hundred runs was made with the thirteen Cuyama burner

installation on this boiler. Curves of Figure 11 show a few of the most important results secured during operation of this boiler.

The range of rates run during this series so far completed was from 0.251 pound per square foot B. W. H. S. per hour to 1.0 pound per square foot B. W. H. S. per hour. The highest overall (boiler, furnace and burners) efficiency obtained over this range was secured at the quarter pound and was 84.23 per cent—the total equivalent evaporation was 48713 pounds. Overall efficiencies and total equivalent evaporations at the half, three-quarter, and one pound rates were respectively as follows: 0.5 pound, 82.93

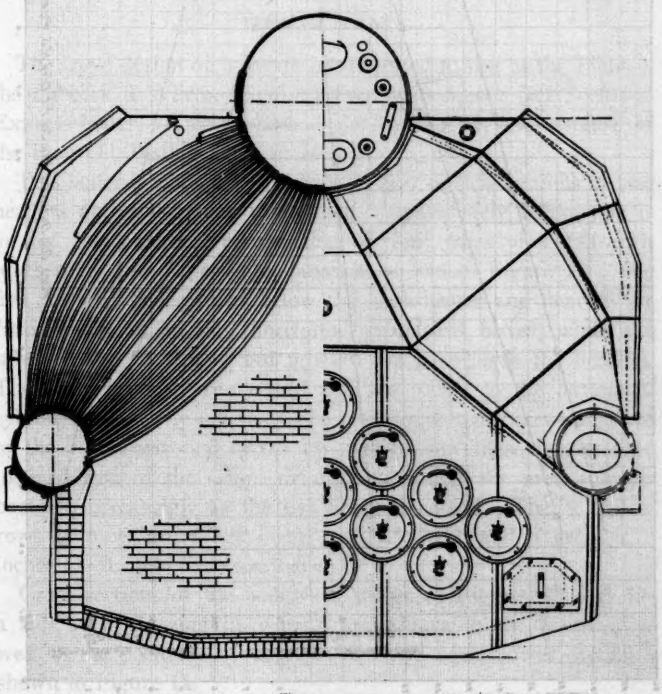


FIGURE 10.

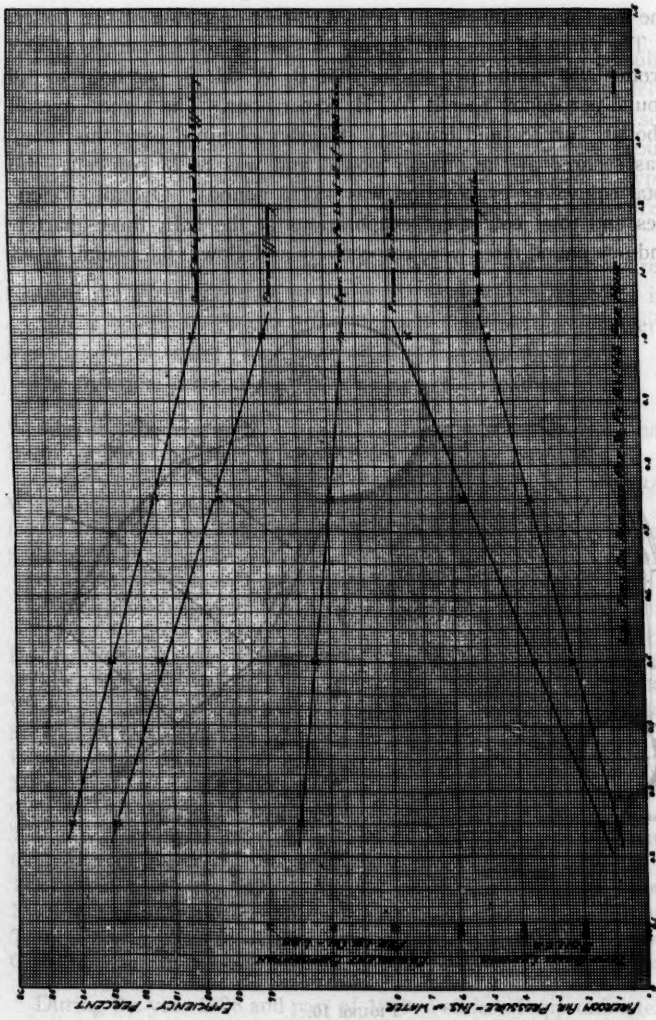


FIGURE 11.

per cent—95,425 pounds; 0.75 pound, 81.57 per cent—140,123 pounds; and 1.0 pound, 80.26 per cent—185,976 pounds.

The fireroom efficiency ranged from approximately 82.9 per cent at the quarter pound rate to 78.0 per cent at the pound. The fireroom air pressure necessary for operation of this boiler with the most efficient number and arrangement of burners at each of the rates run ranged from approximately 1.5 inches at the quarter to about 7.6 inches at the pound; the stack temperatures ranged from approximately 445 degrees F. at the quarter to 640 degrees F. at the pound.

FOURTH BABCOCK & WILCOX BOILER INSTALLED AT THE FUEL OIL TESTING PLANT.

The latest design of boiler to be subjected to test at the Plant is the Babcock & Wilcox High Pressure 5083 square feet Sectional Express boiler. It was completed in April, 1929, and installed at the Fuel Oil Testing Plant in May.

This boiler is made up principally of five vertical sections of two headers and 400–1 inch tubes each. Immediately below this is located the superheater consisting of four rows of 1–1/4 inch tubes rolled into superheater boxes as shown in Figure 12. The five rows of boiler tubes below the superheater and nearest the furnace are rolled into horizontal cylindrical boxes, while the tubes of the sixth row and upward are rolled into the headers. The front headers are vertical and are joined to the horizontal cylindrical boxes by nipples. The rear headers are likewise joined to the rear horizontal boxes by nipples, but these headers are inclined toward the steam drum, constricting the area for gas passage increasingly as the gas becomes cooled. The A and B rows of tubes are 1–1/2 inches O. D.; C, D, and E are 1–1/4 inches O. D.; all others are 1 inch O. D.

Cross sections of the boiler are shown in Figures 12 and 13; a side view showing its general appearance in the fireroom as well as the positions of soot blowers and soot blower piping is shown in Figure 14.

The principal particulars of the boiler are as follows:

Working pressure, pounds	600
Height—floor to top drum over rivets, feet and inches	17–0
Width—over drum manhole fittings, feet and inches	10–7
Length—face of drum feed nozzle to extreme rear, feet and inches	13–6–1/4
Space occupied, cubic feet	1692
Distance from front plate to bulkhead required for tube renewal, feet and inches	10–8
Heating surface 1 boiler, square feet	5083
Superheating surface 1 boiler, square feet	516
Weight 1 boiler dry, pounds	82,250
Weight 1 boiler and superheater, wet (steaming level and 600 pounds working pressure), pounds	89,850
Weight 1 boiler and superheater, wet (steaming level and 400 pounds working pressure), pounds	90,150
Original furnace volume, cubic feet	423.6
Number of steam drums	1
Number of lower drums	2
Number of headers	10
Rows of 1 inch tubes (rolled into headers)	38
Rows of 1–1/4 inch tubes rolled into lower drums	3
Rows of 1–1/2 inch tubes rolled into lower drums	2
Total number of tube rows	43
Number of 1–1/2 inch O. D. tubes, rolled into lower drums	70
Number of 1–1/4 inch O. D. tubes, rolled into lower drums	125
Number of 1 inch tubes O. D. rolled into headers (400 per header)	2000
Total number of B. W. H. S. tubes	2195
Superheater—four rows of 40 tubes each (all 1–1/4 inch tubes)	160
Superheat, degrees F.	200–300

It will be noted from Figure 12 that the burners are shown located on the front or low side of the furnace. But the boiler after its arrival at the Plant was fitted with five Cuyama burners on the rear wall (high wall) of the furnace. Flame from these burners thus travelled into a contracting furnace volume instead of the normally expanding one. More than one arrangement of burners has been used in operating this boiler, but that of the five on the rear wall (high wall) has been used for a complete series. Only results obtained with this five burner arrangement will, therefore, be considered in this article; data secured with other arrangements will be discussed in a future article to appear in this JOURNAL.

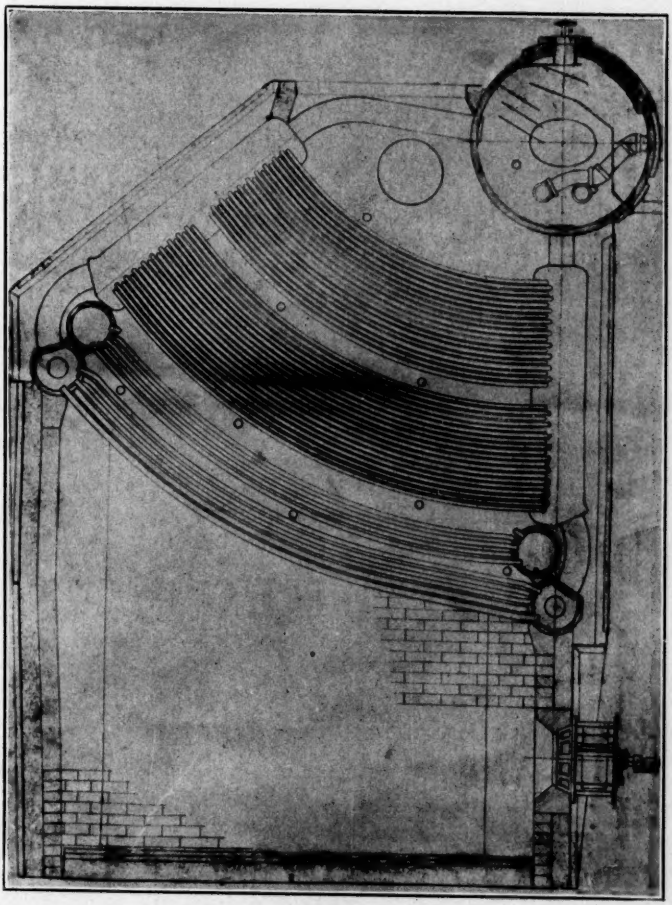


FIGURE 12.

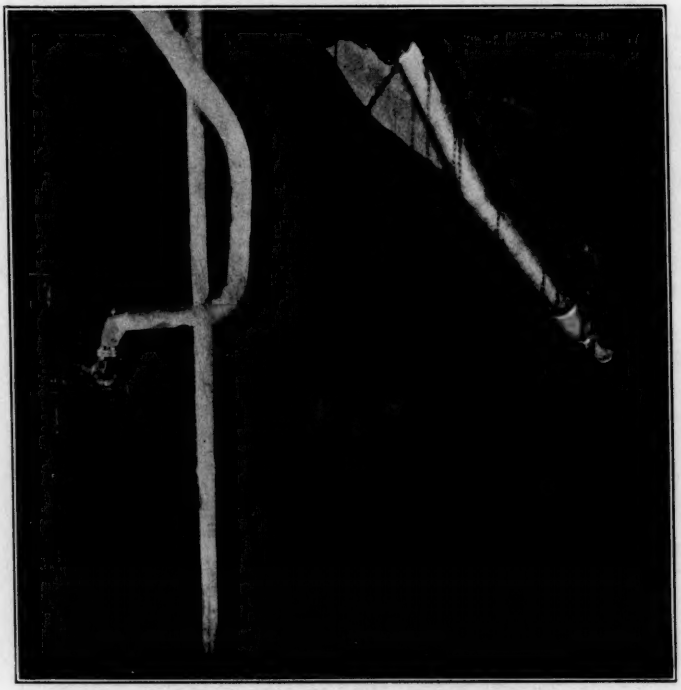


FIGURE 14.
VIEW OF SECTIONAL EXPRESS BOILER.

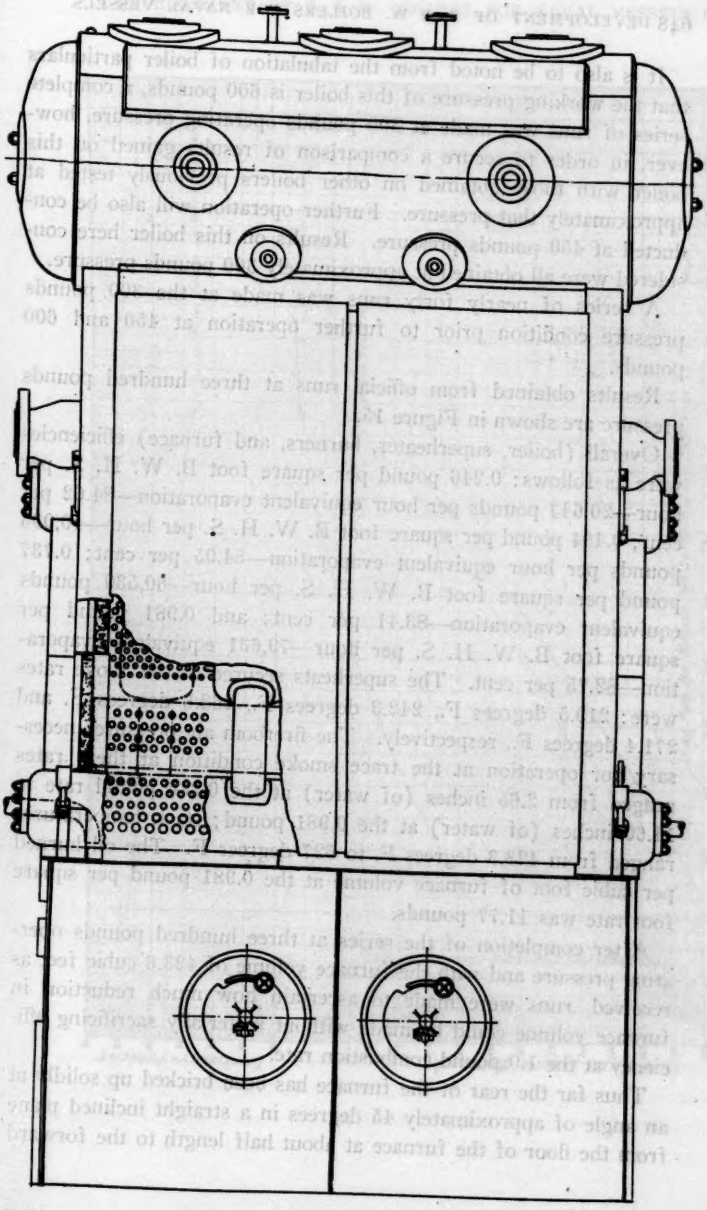


FIGURE 13.

It is also to be noted from the tabulation of boiler particulars that the working pressure of this boiler is 600 pounds, a complete series of runs was made at 300 pounds operating pressure, however, in order to secure a comparison of results gained on this boiler with those obtained on other boilers previously tested at approximately that pressure. Further operation will also be conducted at 450 pounds pressure. Results on this boiler here considered were all obtained at approximately 300 pounds pressure.

A series of nearly forty runs was made at the 300 pounds pressure condition prior to further operation at 450 and 600 pounds.

Results obtained from official runs at three hundred pounds pressure are shown in Figure 15.

Overall (boiler, superheater, burners, and furnace) efficiencies were as follows: 0.246 pound per square foot B. W. H. S. per hour—20,643 pounds per hour equivalent evaporation—84.62 per cent; 0.484 pound per square foot B. W. H. S. per hour—40,095 pounds per hour equivalent evaporation—84.05 per cent; 0.737 pound per square foot B. W. H. S. per hour—60,530 pounds equivalent evaporation—83.41 per cent; and 0.981 pound per square foot B. W. H. S. per hour—79,631 equivalent evaporation—82.75 per cent. The superheats secured at the above rates were: 219.5 degrees F., 242.3 degrees F., 263.6 degrees F. and 271.4 degrees F., respectively. The fireroom air pressures necessary for operation at the trace smoke condition at these rates ranged from 2.65 inches (of water) at the 0.246 pound rate to 11.60 inches (of water) at the 0.981 pound; stack temperatures ranged from 423.3 degrees F. to 527 degrees F. The oil burned per cubic foot of furnace volume at the 0.981 pound per square foot rate was 11.77 pounds.

After completion of the series at three hundred pounds operating pressure and with the furnace volume of 423.6 cubic feet as received, runs were made to ascertain how much reduction in furnace volume could be made without materially sacrificing efficiency at the 1.0 pound combustion rate.

Thus far the rear of the furnace has been bricked up solidly at an angle of approximately 45 degrees in a straight inclined plane from the floor of the furnace at about half length to the forward

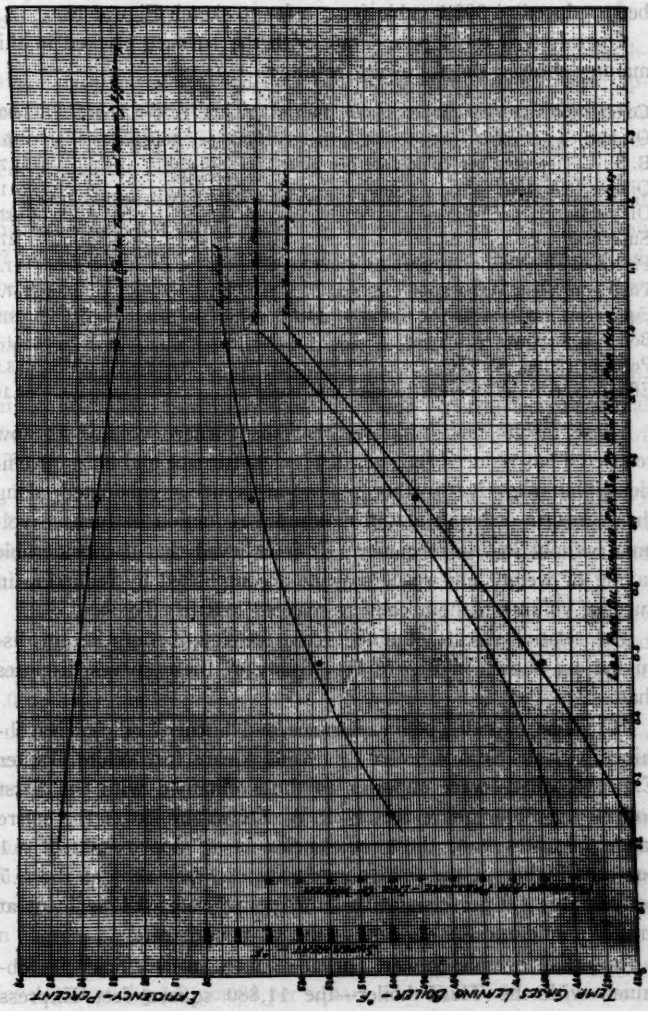


FIGURE 15.

mud drum opposite the burners. The furnace volume thus has been reduced to 302.7 cubic feet; refer to sketch Figure 18.

Operation at the 1 pound rate with the furnace bricked in this manner showed the following results:

Combustion rate pounds per square foot B. W. H. S. per hour	1.006
Oil burned per cubic foot present furnace volume per hour	16.9
B. T. U. liberated per cubic foot per hour	317,179
Oil per hour total	5116
Oil per burner per hour	1023
Superheat	272
Fireroom air pressure (inches of water)	11.9
Total equivalent evaporation	81,237.3
Equivalent evaporation per pound of fuel	15.879
Boiler horsepower developed	2354.7
Per cent rating, per cent	463.2
Efficiency (overall) boiler, furnace and burners	82.10

These results tend to prove that the furnace volume as now restricted has reduced the overall (boiler, furnace and burner) efficiency but slightly; further operation is being conducted along these lines in order to obtain maximum reduction in furnace volume without loss in efficiency. It is believed that a considerable saving of weight and space may be accomplished by reduction in furnace volume without causing marked loss in efficiency.

The relative results obtained from the four boilers herein discussed are shown in Figure 16. Study of these curves discloses that:

(1): The overall (boiler, furnace and burner) efficiencies obtained with the Second boiler—the 7565 square feet White Forster Express boiler—were superior to those obtained with the First boiler—the 4000 square feet B. & W. *Oklahoma*—over the entire range of rates run, these efficiencies being approximately 2.1 per cent higher at the 0.25 pound rate; 3.7 per cent at the 0.5 pound rate; 4.0 per cent at the 0.75 pound rate; 3.4 per cent at the 1.0 pound and 3.0 per cent at the 1.1 pounds rate.

(2): The overall (boiler, furnace and burner) efficiencies obtained with the Third boiler—the 11,880 square feet Express boiler—were superior to those obtained with the First boiler over the entire range of rates run, these efficiencies being approximately

4.4 per cent higher at the quarter, 3.7 per cent at the half, 3.0 per cent at the three-quarter, and 2.8 per cent at the pound.

(3): The overall (boiler, furnace and burner) efficiencies obtained with the Third boiler as compared with Second, were approximately 2.2 per cent higher at the quarter, almost exactly the same at the half; 1 per cent less at the three-quarter and 0.6 per cent less at the pound.

In this connection it is to be noted that, in spite of the fact that this boiler was not equipped with a superheater, and had more than 57 per cent greater boiler water heating surface than the second, the overall efficiencies obtained surpassd those of the 7565 square feet White Forster at rates up to the 0.5 pound per square foot, and were only 1 per cent or less below them at the higher rates.

(4): The overall (boiler, furnace and burner) efficiencies obtained with the Fourth boiler—the 5083 square feet Sectional Express boiler—were superior to those obtained with all three of the boilers previously tested.

They were higher than those obtained on the Third boiler by 0.3 per cent at the 0.25 pound rate; 1.1 per cent at the 0.50 pound rate; 1.8 per cent at the 0.75 rate; and 2.4 per cent at the 1.0 pound rate.

They exceeded those secured on the Second boiler by nearly 2.6 per cent at the 0.25 pound rate; 1.1 per cent at the 0.50 pound rate; 0.7 per cent at the 0.75 pound rate; and 1.8 per cent at the 1.0 pound rate.

And they were greatly above those of the First boiler, viz., nearly 4.7 per cent at the quarter; 4.7 per cent at the half; 4.8 per cent at the three-quarter, and 5.2 per cent at the pound rate.

(5): The superiority in overall efficiency of the Fourth boiler over the other three previously tested is thus aptly demonstrated. A tabulation showing comparative results obtained from the four boilers, including weights and minimum operating space is shown in Figure 17. Study of this tabulation discloses that:

(1): The space occupied by these boilers *per boiler horsepower developed* at the 1.0 pound per square foot B. W. H. S. per hour combustion rate—which is between 450 and 460 per cent rating on these boilers—was: (*a*) 0.86 cubic foot in the case of the Cross drum *Oklahoma* boiler; (*b*) 0.80 cubic foot with the 7565 square

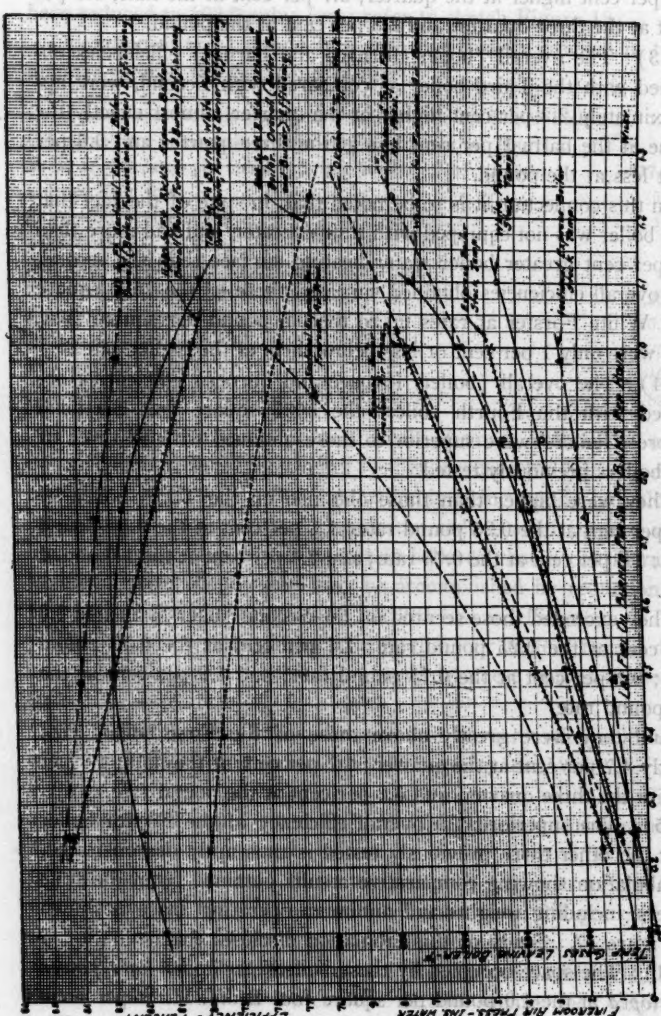


FIGURE 16.

feet White Forster boiler—approximately 7 per cent reduction per B. H. P.; (*c*) 0.73 cubic foot with the 11,880 square feet Express boiler—over 15 per cent reduction per B. H. P. compared with the first boiler and nearly 10 per cent compared with the 7565 square feet White Forster; (*d*) 0.73 cubic foot with the high pressure sectional Express boiler.

COMPARATIVE RESULTS

TYPE OF B.&W. BOILER	COMB. R.T.H. B/SQ. FT.	TOTAL EVAP. STEAM/HR. - POUNDS.	BOILER HP DEVEL.	PERCENT RATING %	OVERALL EFFICIENCY %	CU.FT. OCCUPIED	SQ.FT. OF HEAT. SURF. AT 1/4" THICK.	CU.FT. SPACE PER BOILER HP DEVEL.	SQ.FT. HEATING SURF. BOILER/BOILER HP DEVEL.
4000 SQ.FT. "OLD-STYLE" CROSS-TUBE BOILER	-NAP.	DATA FOR 4000 SQ. FT. BOILER IS ONLY APPROXIMATE							
	0.515	14409	417.7	104.4	80.0	1880	62.820	4.78	127.8
	0.40	25568	741.1	185.3	79.6	"	"	2.13	111.5
	0.65	40612	1177.2	294.3	79.0	"	"	1.34	70.2
8000 SQ. FT. SUPER-HEATER	1.00	63280	1834.2	458.6	77.5	"	"	0.86	45.1
	1.23	75276	2181.9	545.5	78.5	"	"	0.72	37.9
7565 SQ. FT. "WHITE-FORSTER"	0.103	15521	451.6	56.7	81.43	2775	109.800	6.15	243.1
	0.251	30871	896.1	112.1	82.00	"	"	3.16	123.9
WITH SUPER-HEATER	0.502	61947	1795.6	224.6	82.95	"	"	1.56	61.8
	0.753	91762	2680.8	351.7	82.55	"	"	1.04	41.3
	0.858	102798	3010.0	397.9	81.99	"	"	0.92	36.5
	1.00	113298	3460.6	457.4	80.85	"	"	0.80	31.7
	1.11	131312	3810.0	503.6	80.04	"	"	0.73	28.8
EXPRESS WITH 11 DRUMS	1.501	171971	4985.0	659.0	76.15	"	"	0.56	22.0
11880 SQ.FT. H.P. EXPRESS WITH SUPER-HEATER	0.251	48713.1	1412.0	118.6	84.33	2987	152300	2.78	97.3
	0.499	95424.6	2765.9	232.6	82.95	"	"	1.42	44.8
	0.750	140125.0	4061.9	341.9	81.57	"	"	0.97	30.3
	1.00	189276.0	5390.6	453.7	80.24	"	"	0.75	23.9
9063 SQ.FT. SECTIONAL EXPRESS WITH SUPER-HEATER	0.246	30643.0	890.4	117.7	84.59	1692	90300	2.80	180.9
	0.484	60095.4	1162.2	228.6	84.08	"	"	1.46	77.7
	0.737	80339.9	1764.8	346.1	83.41	"	"	0.96	51.5
	0.981	78580.7	2308.1	454.1	82.75	"	"	0.72	39.1

FIGURE 17.

It is not to be overlooked that the space occupied by the last boiler can be further materially reduced by reduction of furnace volume along the lines described and by the building of larger units.

(2): The weight of these boilers *per boiler horsepower developed* at the 1.0 pound per square foot B. W. H. S. per hour combustion rate—was: (*a*) 45.1 pounds in the case of the cross drum *Oklahoma* boiler; (*b*) 31.7 pounds with the 7565 square feet White Forster—approximately 29.9 per cent reduction per B. H. P.; (*c*) 30.3 pounds with the 11,880 square feet Express boiler 32.8 per cent reduction compared with the first boiler and 4.4 per cent compared with the second; and 39.1 pounds with the high pressure Express boiler built for 600 pounds steam pressure. It is to be noted, that in spite of the stronger construction necessary for this higher pressure, this weight per B. H. P. of the sectional

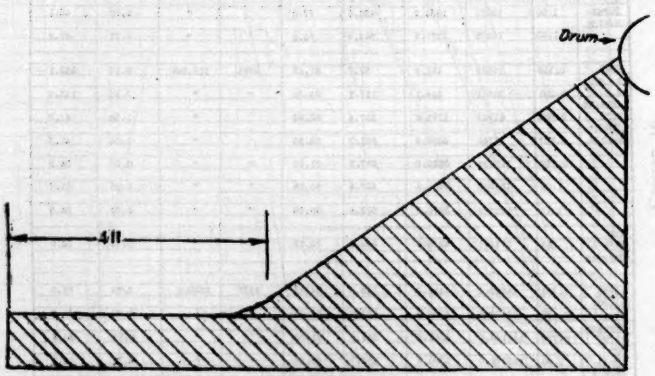


FIGURE 18.

Express boiler is 13.3 per cent per B. H. P. less than the original cross drum *Oklahoma;* furthermore, the reduction of furnace volume along the lines described, will materially reduce this figure.

The weights of the Sectional Express and the three drum Express type boilers of the same size, both built to operate at 300 pounds working pressure, are approximately the same—but if the three drum type were built to operate at 600 pounds pressure, its weight for the same size of unit would be much greater than the sectional Express type built to operate the same pressure.

From the foregoing it will be seen that the limit in possible efficiencies under present Naval fireroom conditions is being gradually approached, it is believed, however, that we are not yet near the possible limit for reduction in space and weight for the amount of steam generated without loss of safety or efficiency.

NOTE: A detailed discussion by Mr. Brierly embodying further results on the High Pressure Sectional Boiler will appear in an early issue of the JOURNAL.

LUBRICATING OIL FOR THE NAVY.

By Comdr. M. C. Bowman, U. S. Navy, Member.

The prime requisite for Naval Machinery is reliability. In order that every bit of apparatus will function efficiently under the demands of full power during the hectic moments of battle, each piece must be of proper design and material, and every unit must be properly assembled and maintained. If the materials of design must be of highest quality to satisfy the conditions imposed upon them, then it would be folly for the materials of maintenance to be less than the highest quality obtainable. Lubricating oil is the most important material of maintenance and the responsibility in providing oils of highest quality is positive.

In discussing the Navy's methods in choosing oils for Naval use the prime consideration is that satisfactory oils cannot be obtained on the basis of any written specifications alone that have been evolved to date. The assertion is not made that satisfactory written specifications cannot be drawn up, because engineering achievement seems to have no limit. But since no satisfactory specifications have been evolved to date, the Navy in discharging its responsibility in the operation of its vessels, has proceeded along practical lines and accepted that the surest proof of the satisfactoriness of an oil comes from its use in actual service.

The instances where the purchases of lubricating oils were made on a basis of specifications have led to operating difficulties directly traceable to the quality of the oil received. Since 1865 there have been tentative specifications for various oils to which as a final test, have been added requirements for actual service tests. These tentative specifications were not kept up to date until 1899, when they were revised but were not followed in the purchasing of oils. Instead a few known acceptable brands were purchased on a proprietary basis which established a monopoly for three

firms which was displeasing to their rivals and who prevailed upon the Department to direct that oils be bought strictly on a specification basis. Such an order was promulgated in April, 1905. At about this time, the armored cruiser class of vessels were being added to the fleet. Forced engine lubrication came into established use on these vessels and the demand for a high grade of mineral engine oil was not met under the system of purchase. Here is quoted a memorandum from the Bureau of Steam Engineering to the Department dated 17 November 1906, which is in substance the attitude of today, an exception being noted to the statement in paragraph 5 that mineral oil alone is not a good lubricant:

MEMORANDUM FOR THE DEPARTMENT.

SUBJECT: ENGINE LUBRICATING OIL.

"1. The Bureau of Steam Engineering wishes to bring to the Department's attention the unsatisfactory condition of affairs in the Engineer Department of most of the ships of the Navy, due to the enforced use of inferior brands of engine lubricating oil.

2. The enforced use of inferior oil has been brought about by the Department's order forbidding the purchase of engine lubricating oil by brand name, as had been the custom for many years, and directing the purchase of oil to specifications.

3. As a general rule the purchase of supplies by open competition, subject to specifications, and under certain well defined restrictions, is the right thing to do; but the Bureau believes that there are some supplies that, to be satisfactory, will have to be furnished by those manufacturers who have made a study of the peculiar and special conditions such supplies are intended to meet; and who, either by means of the control of certain sources of supply, or by improved methods of manufacture, or by both, are able to furnish exactly the article needed, in any desired quantity, and of an unvarying quality. In this Bureau's opinion engine lubricating oil for naval machinery is an exception to the above mentioned general rule, and the following is offered in support of this belief.

4. The peculiar and special conditions mentioned are found in our naval engines because no other machinery has to support the pressures per square inch of bearing surfaces that are necessary when running under full power conditions. Our war ships exist for these full power conditions, and to supply them with a lubricating oil that fails under those conditions of service would be as fatal as to furnish the ships with guns not strong enough for full charges of powder. The design of the machinery to fit in the narrow limits imposed by war ship conditions requires the use of the very best materials of construction and the very best materials for operation.

5. The Bureau is of the opinion that the difference between a successful engine lubricating oil for high class machinery, and one that is not satisfactory, lies as much in the skill with which the necessary elements are compounded as in their quality. It is well known that mineral oil alone is not a good lubricant; but, when used as a base or carrier for the fixed oils of better lubricating qualities, the resulting compound may be a highly successful lubricant. Much depends upon the proper grade of the base oil, and of the fixed oil or oils. Much also depends upon the amount of refining the base oil is subjected to, and also much depends upon the skill with which the compounding is done. It is obvious that it is these last two requirements of a successful compounded oil that elude the closest and most carefully drawn specifications. It is also certain that specifications do not assure the obtaining of oils of uniform quality, a very important requirement, and one not to be had except from manufacturers fitted out for this special work.

6. Owing to the limited storage room for oil available at most of our Navy Yards, small purchases of oil are the rule, and this fact alone makes it so much more necessary to have one or more large sources of supply from which to draw desired quantities of lubricating oil of an assured quality. This condition cannot obtain when the purchases that are made are in the main from such dealers or refiners as have but a limited capacity for supplying the grade of oil needed. It is, unfortunately, these small dealers or refiners, who are most eager to supply a cheap oil to the Navy; and, so far as this Bureau can learn these refiners are the very ones who cannot command sources of supply known to be the best

for engine lubricating oil. This economic fact, however unfortunate it may be, must not be ignored when considering the purchase of an article that must be of the very best quality.

7. The net result of eighteen months of earnest effort is that no warship in commission has been able to make a successful full power trial with a specification bought oil. Many attempts have been made to do so, with a result disastrous to the machinery, entailing a cost for repairs to bearings out of all proportion to the apparent gain in the first cost of the cheaper oil. The words "apparent gain" are used advisedly, because it is notorious that the cheaper oils cost more in the end on account of the increased amount required to do the work, without counting the extra cost of repairs.

8. As a purely business proposition, the purchase of oil by specifications is not a success. In connection with this subject, the Bureau invites attention to the reports sent in by the Commander-in-Chief of the Atlantic Fleet, and by others. The extra repair bills alone mount into the thousands of dollars, to say nothing of the loss of efficiency of the ships.

9. It is also certain that no contractor has ever attempted to run a full power trial of a Government vessel with any one of the several brands of oil now being purchased for issue to our ships.

10. The total result then of the effort to buy oil by specifications is that no ship of the fleet supplied with inferior oil, that is the result of open competition can satisfactorily make a full power run. Repair bills amounting to thousands of dollars have been incurred that are directly attributable to the use of poor oil. Several failures of late of new ships to repeat after six months commission the acceptance trial performance as to power and speed are, in this Bureau's opinion, the result of using inferior oil. Taking no account of the cost of repairs due to its use, it is certain that the money cost of the inferior oil is greater. That the Ships of the Fleet have been able to get along as well as they have done is due to the fact that they were able when away from a Navy Yard to buy a supply of well known and approved oil. This source of relief from a hard situation will be to a large extent removed with the issuing of the Department's proposed special order concerning the purchase of engine lubricating oil away from Navy Yards."

Difficulties continued until February, 1907, when the President after a visit to the fleet where he learned of the trouble due to burned out bearing brasses, directed the Secretary of the Navy to have the Engineer-in-Chief report on the situation direct to the White House. The Engineer-in-Chief replied under date of 17 April 1907, in part as follows:

"3. Within the past two years, and not before complaints about the anti-friction metal used in our large and high powered ships have been loud and frequent. In this same space of time the lubricating oils, formerly successfully used in the Navy, have very generally been replaced by cheaper and inferior grades, due to the new system of buying oils by specifications as distinguished from the method of buying by brand name. The two conditions were coincident; that is to say, reports of the failure of the anti-friction metal began to come in at the very time the change in the lubricating oil used was taking effect."

Coincident with the above report, the Secretary of the Navy issued an order to the Bureau of Steam Engineering in substance directing that a list of various brands of lubricating oils and other stores as packings, fire-brick, tool-steels, etc., that have been found satisfactory for Naval use be furnished. Requisitioning for this material on a proprietary basis was authorized. The Bureau of Steam Engineering responded by naming 4 brands of lubricating oil for main engines, 3 brands for high speed engines, 3 brands for ice machines, 6 brands for gas engines, and 1 brand for turbines. Apparently this marks the beginning of the Acceptable List System.

Another quotation from a memorandum of the Bureau of Steam Engineering dated 12 February 1907 is as follows:

"For the past two years the oils used in the Navy have been purchased in the open market subject to specifications, but the plan has not worked to the interest of the Government, especially for engine lubricating oils. Previously the oils were purchased by brand-name, without specifications, from *manufacturers making a special study of the Navy's needs in lubricating oils.* That plan gave satisfaction, but it necessarily limited the purchase to a few oil firms of large resources." Referring to the italicised words above, it was soon found necessary for the *Navy* to make

the special study of the Navy's needs and to select the brands. The number of suppliers who have entered into the field now have become large enough to provide satisfactory competition and unrestricted quantities. The Engineering Experiment Station at Annapolis, upon its establishment in 1910, undertook this study and has been charged with all matters of development and testing since that time. The cooperation of all suppliers with that station has been of the highest and most profitable order.

In the above paragraphs stress has been laid on conditions about 1905 to 1907 because that period marks the beginning of the development of strictly mineral oils for forced feed and motor cylinder oils which include oils for forced feed engines, both reciprocating and turbine, and for internal combustion engines. The grades of oils named on the Annual schedules from those years on indicate the beginning of demands for oils created by new types of machines all using forced feed lubrication. Since 1918, the only new demands of any importance have been created by development in aircraft. Shortly after the world war the diversity of types of engines tended to increase the grades of oils but latter day design has become more standardized so that the tendency has been to reduce the number of grades. Within the last few years machine types have tended toward stabilization and design has approached standardization so that in the Navy it has been possible to take stock and to survey the lubrication problem by defining a category of grades of lubricating oils which will fulfill all requirements. The purpose of lubrication is to exchange the friction between two relatively-moving metal surfaces for the far lesser friction of the film of oil which must persist between these surfaces under bearing pressures, moving speeds and temperatures. The elements of friction have not changed, therefore lubrication requirements have changed and will be changed in degree only. The lubricating oil industry grew up by developing a brand of oil for every demand until the number of brands numbered from 350 to 400, a most uneconomical situation. Such a condition was confusing and unsound. A survey of the problem led the Navy to an attempt to standardize on as few grades as possible and to classify its needs on a rational basis.

The machinery of the Navy was divided into categories of operation which resolved into four as follows:

(1) Refrigeration.
(2) Forced-feed and Motor cylinders.
(3) Steam Engine.
(4) Air-compressor.

Each of these categories of operation was examined to determine its scope and 8 classes of oils in the categories were established covering service in more detail. Each class was then divided into grades, covering the viscosity ranges required. Each grade was called an item, of which there are 17. Each item is indicated by a four digit symbol developed as follows:

A four-digit identifying symbol is used in all cases, the first digit indicating the class of oil and the last three digits indicating the viscosity. Viscosity at 130 degrees F. is used for refrigeration oils, for forced feed and motor cylinder oils, except aviation oils, for compounded air-cylinder oils, and for transformer oils. Viscosity at 210 degrees F. is used for aviation oils, for compounded marine engine and cylinder oils, and for mineral marine engine and cylinder oils.

The classes of oils indicated by the first digit are as follows:

1—Refrigeration oils.
2—Forced-feed and motor-cylinder oils.
3—Forced-feed and motor-cylinder oils (aviation).
4—Compounded marine engine oils.
5—Mineral marine engine and cylinder oils.
6—Compounded steam-cylinder oils (tallow).
7—Compounded steam-cylinder oils (lard or tallow).
8—Compounded air-cylinder oils.
9—Transformer oils (not a lubricant).

The following examples will illustrate the symbol system:

(*a*) 2135 (that is, two one thirty-five) indicates a force feed and motor cylinder oil having a viscosity approximating 135 seconds at 130 degrees F.

(*b*) 3080 (that is, three zero eighty) indicates an aviation oil having a viscosity approximating 80 seconds at 210 degrees F.

(*c*) 4065 (that is, four zero sixty-five) indicates a compounded marine-engine oil having a minimum viscosity of 65 seconds at 210 degrees F.

In the following tabulation column 1 gives the symbol designation, column 2 the old designations, columns 3 and 4 and the viscosity.

Symbol	Kind	Viscosity at 130 F. (seconds)	Saybolt at 210 F. (seconds)
Refrigeration Oil:			
1075	Ice machine	65–85	.....
Forced-Feed and Motor Cylinder Oils			
2110	Light	100–115	.....
2135	Medium	125–145	.....
2190	Heavy	180–200	.....
2250	Extra heavy	240–260	.....
2310	Ultra heavy	300–320	.....
3080	Aviation, winter	.....	75–85
3100	Aviation, summer	.....	90–100
3120	Aviation, heavy	.....	115–125
Engine Oils			
4065	Marine engine compounded		Min. 65
5065	Mineral marine engine		Min. 65
5150	Mineral steam cylinder		135–165
5190	Mineral steam cylinder, superheat		160–220
6135	Compounded steam cylinder		120–150
7105	Compounded steam cylinder		95–110
Air Compressor Oils			
8135	Compounded air cylinder	125–145	.....
8190	Compounded air cylinder	180–200	.....
Electrical Oil			
9045	Transformer oil	40–45	.....

NOTE: For purposes of development and purchase, an additional item 18 for transformer oils has been added. It is not a lubricant.

Having defined the lubrication scope it then became necessary to determine what oils would fill the requirements. In order that no oils will be purchased that are not known fully to meet service requirements, an Acceptable List is maintained on which is listed those oils which are positively known to be acceptable. No offering of an oil not on this list is entertained.

ACCEPTABLE LIST.

In order to determine the acceptability of a lubricating oil submitted as a definite item as classified above, the Bureau of Engineering conducts laboratory tests at the Engineering Experiment Station at Annapolis, Md. In addition to the laboratory tests, service tests are conducted as follows: Symbols 1075, 2110, 2310, 3080, 3100 and 3120 on equipment of the Station; Symbols 2135, 2190, 2250, 4065 and 5065 on shipboard under routine operation. Service tests for Symbols 5190, 6135, 7105, 8135, 8190 and 9045 are not required because these are non-recoverable oils, used but once. If the laboratory tests and the service tests are satisfactory, the name of the manufacturer and the trade or brand name of the oil tested are placed on the "Acceptable List" under proper symbol number of the classification there to remain as long as quality of oil delivered maintains the standard of the sample approved or until it appears that the brand does not embody improvements in requirements adopted since test, in which case approval becomes tentative pending a re-test. Trade or brand name of an oil is regarded as a permanent description representing characteristics and qualities as determined by test.

LABORATORY TESTS.

A complete laboratory test consists of three parts, chemical, physical, practical.

Chemical analysis and physical test—all tests on lubricating oils are made in accordance with standard or tentative standard methods of the American Society for Testing Materials and/or U. S. Government standard test methods shown in Federal Specification No. 2. Tests for SO_3 as sulphonates, unsaturated compounds, ash and evaporation loss are made by methods evolved

TABLE I.—MINIMUM REQUIREMENTS

	Refrigeration; Ice Machine	Forced-Feed and Motor-Cylinder Oils							
		Light	Medium	Heavy	Extra-heavy	Ultra heavy	Winter aviation	Summer aviation	Heavy aviation
Symbol	1075	2110	2135	2190	2250	2310	3060	3100	3120
Color	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)
Reaction	Neutral	Neutral	Neutral	Neutral	Neutral	Neutral	Neutral	Neutral	Neutral.
Neutralization number, maximum	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10.
Fixed or fatty oils (per cent)	None	None	None	None	None	None	None	None	None.
Precipitation number, maximum	None	None	None	None	None	None	None	None	None.
Free sulphur or corrosive compounds	None	None	None	None	None	None	None	None	None.
SO_3 as sulphonates	None	None	None	None	None	None	None	None	None.
Unsaturated compounds (per cent), maximum	6.00	10.00	12.00	14.00	15.00	18.00	30.00	30.00	30.00.
Water	None	None	None	None	None	None	None	None	None.
Carbon residue (per cent), maximum	0.50	0.50	0.50	0.50	1.00	1.00	1.50	1.50	1.50.
Nature of carbon residue	Loose, flaky.	Loose, flaky.	Loose, flaky.	Loose, flaky.	Loose, flaky.	Loose, flaky.	Loose, flaky.	Loose, flaky.	Loose, flaky.
Ash (per cent), maximum	None	None	None	None	None	None	0.01	0.01	0.01.
Total sulphur (per cent), maximum	0.50	0.50	0.50	0.50	0.50	0.50	0.75	0.75	0.75.
Evaporation loss (per cent), maximum	4.00	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50.
Nature of the oil	Mineral	Mineral	Mineral	Mineral	Mineral	Mineral	Mineral	Mineral	Mineral.
Specific gravity at 60/60° F	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)
A. P. I. degrees	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)
Flash point, minimum	310° F.	325° F.	335° F.	345° F.	355° F.	365° F.	390° F.	410° F.	450° F.
Pour point, maximum	0° F.	35° F.	35° F.	35° F.	35° F.	35° F.	15° F.	25° F.	30° F.
Viscosity, Saybolt, Universal:									
At 130° F. (seconds)	65–85	100–115	125–145	180–200	240–260	300–320			
At 210° F. (seconds)							75–85	90–100	115–125.
Emulsion test (time to settle out (minutes), maximum):									
Distilled water at 130° F	30	30	30	30	30				
1 per cent salt solution at 130° F	30	30	30	30	30				
Distilled water at 180° F									
1 per cent salt solution at 180° F						60	60	60	60.
Normal NaOH at 130° F						60	60	60	60.
Boiling water at 120° F	30	30	30	30	30				
Emulsifying constituents with steam	30	30	30	30	30				
	None	None	None	None	None	None	None	None	None.

TABLE I.—CONTINUED.

	Engine Oils								Electric Transformer	Methods of Tests	
	Compounded engine	Mineral engine	Mineral steam cylinder		Compounded steam cylinder		Compounded air cylinder				
Symbol	4065	5065	5150	5190	6135	7105	8135	8190	9045	A. S. T. M. No.	Federal Specification Board No.
Color	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)	D155-21T	10.2.
Reaction	Neutral	Neutral	Neutral	Neutral	Neutral	Neutral	Neutral	Neutral	Neutral		510.1.
Neutralization number, maximum	3.00	1.00	0.15	0.15	0.40	0.50	1.00	0.25	0.10	D126-27T	510.21.
Fixed or fatty oils (per cent)	10-20	None	None	None	5-7	9-10	2-4		None	D94-21T	548.1.
Nature of fixed or fatty oils	Preferably blown rapeseed.				Tallow oil	Tallow or lard oil.	Lard oil	Lard oil			
Precipitation number, maximum	None	None	0.05	0.05	0.05	0.05	None	None	None	D91-21T	510.1.
Free sulphur or corrosive compounds	None	None	None	None	None	None	None	None	None		530.21.
SO_3 as sulphonates	None	None	None	None	None	None	None	None	None		See paragraph 16a.
Unsaturated compounds (per cent), maximum									4.00		See paragraph 16b.
Water	None	None	None	None	None	None	None	None	None	D95-27	508.12.[2]
Carbon residue (per cent), maximum			4.50	4.50	2.50	2.00	0.50	0.50	0.02	D189-27	508.12.
Nature of carbon residue			Loose, flaky	Loose, flaky	Loose, flaky	Loose, flaky	Loose, flaky	Loose, flaky	Loose, flaky		
Ash (per cent), maximum			0.05	0.05	0.05	0.05	None	None	None		See paragraph 16c.
Total sulphur (per cent), maximum	0.50	0.75	0.50	0.50	0.50	0.50	0.50	0.50	0.50	D129-27	530.21.
Evaporation loss (per cent), maximum		2.50	Trace	Trace	Trace	Trace	2.50	2.50	3.00		See paragraph 16d.
Nature of the oil	Compounded	Mineral	Mineral	Mineral	Compounded	Compounded	Compounded	Compounded	Mineral		
Specific gravity at 60/60° F	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)		By pykneometer.
A. P. I. degrees	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)	(1)		Circular, Bu. Stds., No. 154.
Flash point, minimum	360° F.	345° F.	490° F.	525° F.	475° F.	450° F.	325° F.	245° F.	270° F.	D92-24	110.22
Pour point, maximum	35° F.	35° F.	60° F.	60° F.	60° F.	40° F.	30° F.	30° F.	30° F.	D97-27[3]	30.12.
Viscosity, Saybolt, Universal:											
At 130° F. (seconds)							125-145	190-200	40-45	D88-26	30.41
At 210° F. (seconds)	65	65	135-165	160-220	120-150	95-110					
Emulsion test (time to settle out (minutes), maximum):											
Distilled water at 130° F	60[4]	30[4]							10		330.11.
1 per cent salt solution at 130° F	60[4]	30[4]							10		
Distilled water at 180° F											330.21.
1 per cent salt solution at 180° F											
Normal NaOH at 130° F									10		330.11
Boiling water at 130° F									10		
Emulsifying constituents with steam									None	Conradson steam emulsion test.	
Wick feed tests	(1)	(1)									509.1.
Dielectric strength									23,000 volts	D117-27	

[1] No minimum requirement. In each case determination is made as a matter of record.
[2] Also observation at time of taking flash point.
[3] Oil shall remain emulsified for the number of minutes stated.
[4] Retain 30 per cent flow at end of 14 days.

by the Engineering Experiment Station. Table I outlines certain minimum requirements which must be met before the oil in question will be considered for further test.

Practical tests—these are of three different kinds; for forced feed and motor cylinder, wick-feed tests of marine engine oils and breakdown test of transformer oils.

For forced-feed and motor-cylinder oils tests are made on the Engineering Experiment Station and endurance machine to determine the work value and the relative friction. A test journal 6 inches long and of 3 13/16 inches diameter is lubricated by forced feed. The journal is revolved by an electric motor at a speed of 2000 revolutions per minute and is maintained under a bearing pressure of 150 pounds per square inch. During the test the circulating oil is not cooled but is allowed to attain any temperature it may. The lubricating oil is fed to the upper on-side of the bearing at a pressure of 15 pounds gauge and is continuously circulated from the oil reservoir to the test journal and back to the oil reservoir. The oil is tested in the endurance machine for a total period of 100 hours, readings of the temperatures of the bearings and circulating oil being taken hourly and calculations of the coefficient of friction being made hourly. The lubricating oil is examined chemically and physically after the test to determine its condition and the nature of the changes which have taken place in the oil during the run on the endurance machine.

Wick-feed tests of marine-engine oils is conducted on apparatus and by method 200.1 U. S. Government Specification No. 2.

Transformer oils are subject to dielectric test in accordance with A. S. T. M. method D 117-27.

SERVICE TESTS.

After satisfactorily passing the laboratory tests, the following oils are subjected to service test under operating conditions on board ship:

Symbol 2135 (medium) forced feed and motor cylinder oils.

Symbol 2190 (heavy) forced feed and motor cylinder oils.

Symbol 2250 (extra heavy) forced feed and motor cylinder oils.

Symbol 4065 (marine engine, compounded).

Symbol 5065 (marine engine, straight mineral).

The requisite quantities for service test on board ships varying from 200 to 1000 gallons, depending on the type of machinery and forced feed systems of the vessel conducting test, are furnished the designated vessels by the Bureau of Engineering, who use this oil for routine use paralleling an oil previously proven to be satisfactory for comparative purposes.

Service tests of the following oils are conducted at the Experiment Station:

Symbol 1075 tested in station ice machine.
Symbol 2110 tested in station motor generator.
Symbol 2310 tested in station Diesel engine.
Symbols 3080, 3100, 3120 tested in high compression aviation motor.

WORK FACTOR.

Obviously to accept the lowest bid price on a quantity basis is unsound where high quality of service is necessary, the lowest bid price of a quality unit being the sound element of consideration. In order to establish a basis of comparison between oils as to quality, the Bureau of Engineering has developed and adopted, for use, with forced feed and motor cylinder oils, a work factor which evaluates numerically the deterioration of a given oil through oxidation after 100 hours on the endurance machine. The development of this work factor is aptly described in an article written by Commander Keleher and Mr. McGeary, both of the Bureau of Engineering and published in "Manufacturing Industries" in July, August and September of 1928.

Since the publication of that article, the work factor has been revised by changing the weights allowed some of the factors.

In determining the acceptability of forced-feed and motor-cylinder oils "work values" are used to make comparisons between the following physical and chemical properties after the 100-hour endurance test and those obtained on the same oil before the endurance test:

(*a*) Viscosity.
(*b*) Neutralization number.
(*c*) Precipitation number.
(*d*) Carbon residue.

(*a*) *Viscosity*—Viscosity is determined at three temperatures, viz: 100 degrees, 130 degrees and 210 degrees Fahrenheit. An increase of more than 20 per cent on oils for Symbols 2110 to 2310, inclusive, and of 25 per cent on oils for Symbols 3080 to 3120, inclusive, at any one of the three temperatures marks the oil as not acceptable. Reducing the increase in viscosity to a work-factor basis, zero increase warrants a work value of 1.00 and an increase between zero and 20 or 25 per cent some value less than 1.00. Formula 1 is used to compute the work value of increase in viscosity at each temperature.

(Formula 1)

$$\text{Work value} = 1.00 - \left\{ \frac{\text{Final viscosity} - \text{initial viscosity}}{\text{Initial viscosity}} \right\} \div C$$

C = 0.20 for Symbol 2110 to 2310, inclusive, oils.
C = 0.25 for Symbol 3080 to 3120, inclusive, oils.

(*b*) *Neutralization number,* precipitation number and carbon residue will all increase in content to some extent during the endurance test, the poorer oils showing greater increases. The following tabulation shows the initial acceptable content, the initial rejection content, and the final rejection content for these factors:

Factor	Maximum content initially acceptable	Initial rejection point	Final rejection point
Neutralization number	0.10	0.11	1.01
Precipitation number	None	.05	1.01
Carbon residue, 2110, 2135, 2190 per cent	.50	.55	1.05
Carbon residue, 2250, 2310 per cent	1.00	1.05	2.05
Carbon residue, 3080, 3100, 3120 per cent	1.50	1.55	3.05

Any content beyond that allowable is considered sufficient reason for rejection of an oil. Two ratings are given to the factor carbon residue, one based on the initial content (Formula 2) and the other on the increase in content during the endurance test (Formula 3). The formulas follow:

(Formula 2)

$$\frac{\text{Initial rejection point} - \text{initial content}}{\text{Initial rejection point}} = \text{Work value No. 1.}$$

(Formula 3)

$$\frac{\text{Final rejection point} - \text{final content}}{\text{Final rejection point} - \text{initial content}} = \text{Work value No. 2.}$$

One rating only will be given for neutralization number and precipitation number based on the initial content, which always must be 0.10 or less and zero, respectively, and the increase in content during the endurance test. Formula 4, as follows, will be used:

(Formula 4)

$$\frac{\text{Final rejection point} - \text{final content}}{\text{Final rejection point} - \text{initial content}} = \text{Work value.}$$

After completing the calculations the work factor is developed as follows:

(*a*) Average the three results obtained by Formula 1 for viscosity.

(*b*) Average the results obtained by Formulas 2 and 3 for carbon residue.

(*c*) Collect the different work values as follows:

Viscosity .....
Neutralization number .....
Precipitation number .....
Carbon residue .....

Total .....
Average work value .....

To be acceptable an oil must not exceed any of the "rejection points" shown above and must have an "average work value" of not less than 0.50 after the 100-hour endurance test.

A high work factor has been proven by repeated tests to indicate a high degree of refining. Refining costs money but pro-

duces high quality. Therefore the work factor is applied to purchase to give a service cost as follows:

$$\text{Service cost per gallon} = \frac{\text{Bid price per gallon}}{\text{work factor}}$$

The lowest service cost per gallon is considered to be the low bidder.

The work factor idea is not a new one, it simply is given in a new form. Going back into Navy records, we find a report of a board convened at the New York Navy Yard for testing oils under various dates in 1884. Oils were tested on a friction machine and as shown in Table II evaluated relatively on a basis of "work done in footpounds per Troy grain of lubricant to reach the temperature 220 Fahrenheit of Journal." In another report, as shown in Table III, evaluation was made on the basis of "Footpounds of work done per minute per Troy grain of oil consumed."

The friction or, as now called, endurance machine was used as far back at 1866. A report of a board convened at New York Navy Yard for testing various oils under date of 6 December 1870 described the testing machine as follows: "The machine used for the purpose was constructed by a previous board in 1866. It consisted of a wrought iron shaft with a journal 9 1/2 inches in diameter and 14 inches long at each end, geared to a small oscillating engine. The weight was applied to the journals by means of a compound lever, so arranged that the pressure on the journals could be graduated at pleasure and accurately estimated.

The oil to be tested was supplied to the journals from an oil cup, fixed on the top of each cap brass, through small glass syphons, which had been previously graduated to feed a certain quantity of oil in a given time, according to the fluidity of the oil to be tested. A counter was attached to the shaft by which the number of revolutions were kept, uniformly at 75 per minute.

Small holes were drilled in the end of each cap brass, extending to within a quarter of an inch of the journals, whereby the temperature of the journals could be ascertained at all times."

Each specimen of oil was tested for nine consecutive hours.

The test data given in Tables II and III under date of April, 1884, was gathered from a machine described as follows: "The

Name of the Lubricant.	Duration of the Experiment in consecutive minutes.	Number of Revolutions made by the Journal. Total.	Ditto per minute.	Number of feet passed over by the Journal during the Experiment.	Temperatures in degrees Fahrenheit. Mean of the Journal.	Mean of the Air surrounding Journal.	Maximum of the Journal.	Minimum of the Journal.	Increase of Journal Temperature during Experiment.	Mean incident Load on the centre of the Journal in pounds. Total.	Ditto grams laid of Lubricated perpendicular of Journal.	Mean pull on the dynamometer in pounds.	Coefficient of Friction.	Weight of Lubricant consumed in Troy grains.	Work done in foot-pounds per Troy grain of Lubricant to reach the temperature 220° Fahrenheit of Journal.	Work done in foot-pounds per minute per Troy grain of Lubricant to reach the temperature 220 Fahrenheit of Journal.	Relative commercial values of Lubricants being the quantities in the preceding column expressed proportionally.
Commercial Sperm Oil.	125	23010	184.080	13730.427	139.96	57.2	220	55	162	2007.220	172.603	72.6258	.0376	72	3101225.9270	24809.9670	100.00
Capital Cylinder Oil.	128	23055	180.117	13767.891	143.90	55.8	220	56	162	1732.073	148.941	74.7701	.0418	72	2613659.2405	22249.2621	85.68
Eldorado Engine Oil.	121	22625	186.983	13497.552	147.22	58.5	220	68	152	1389.213	113.676	75.6570	.0553	9	2068806.7228	17097.5762	68.92
Eclipse Grease.	122	21970	180.079	13166.942	144.32	60.0	220	62	158	1442.055	120.172	69.1373	.0477	72	2120134.6393	17378.1511	70.04

TABLE II.

Kind of Oil.	Duration of the Experiment in minutes.	No. of Revolutions made by the Pendulum. Total during the Experiment.	No. of Revolutions made by the Pendulum. Per minute.	No. of feet Retarded by circumference of pendulum during the experiment.	Temperatures in degrees Fahr. Mean of Pendulum.	Mean of Air.	Pendulum. Minimum.	Pendulum. Maximum.	Pendulum. Extremes.	Resistance, deduced from number of pounds in pounds. Mean.	Resistance, deduced from number of pounds in pounds. Sum of the two last horizontal projections.	Ball of the Pendulum, in pounds.	Coefficient of Friction.	Many points of Oil measured.	Work done in Foot-pounds during the experiment per pound of Oil measured & from the Temperature in the time noted.	Foot-pounds of work done per minute per Troy pound of Oil consumed.
Standard commercial sperm.	133	25900	194.985	15629.760	163.67	57.5	220	55	165	1153.1300	99.6450	57.183	.0477	6	2104466. 3802.	23353. 1721
Standard commercial sperm.	133	25625	192.105	15392.749	156.16	57.0	220	58	162	1052.6015	152.6158	46.0931	.0432	9	2132289. 0517	23580. 7507
Commercial sperm from Kere island. Every yard. A	127	22570	193.077	15474.320	156.64	67.0	220	70	150	1640.2307	136.6061	42.2338	.0373	12	2637207. 2295	22556. 4778
B	125	23760	190.080	14938.700	139.90	58.6	220	57	163	1706.5000	162.2333	52.7378	.0309	9	2727778. 3277	21823. 2260
C	115	21122	190.982	14797.110	163.17	68.0	220	70	150	1617.2923	132.2493	57.6972	.0368	10½	2030662. 0113	17657. 3679
D	166	29030	190.363	17090. 918	152.25	66.0	220	66	156	1926.0363	162.2230	56.2276	.0309	9	3466382. 6752	20756. 0578
E	130	24825	190.961	15027. 468	158.55	65.0	220	68	152	1712.3249	152.6072	48.7790	.0312	9	3026478. 7923	23280. 6060
F	131	24995	191.259	15258.321	160.65	65.0	220	68	152	1818.7026	152.6622	41.2385	.0336	9	3070492. 1321	23437. 3293
G	139	26470	190.359	16013. 271	166.31	66.0	220	70	150	1847.0726	156.7902	56.6867	.0302	10½	2847838. 0622	20485. 1659
H	136	26160	192.206	16668.925	157.88	68.0	220	70	146	1782.6623	148.2206	56.2740	.0296	9½	3321122. 7052	24893. 6699
Flax Oil (petroleum)	124	22815	183.990	13912. 880	166.80	67.0	220	67	153	1266.0877	105.5073	52.2200	.0269	8	2286761. 1266	17648. 7296
Cylinderoil (petroleum)	131½	26995	183.935	15132. 307	156.63	75.5	220	70	144	1768.3282	126.2232	57.2091	.0298	9½	2868296. 8060	21609. 6891

TABLE III.

apparatus consisted of a perfectly cylindrical polished chrome steel journal of 2 19/64 inches diameter and 5 13/16 inches length, revolving in stationary bronze bearings composed of 88 parts copper, 10 parts of tin and 2 parts of lead. The oil was supplied to the journal through two holes in the upper bearing—one near each end—and distributed equally over the journals by channels in the bearings. A cavity sufficient to easily contain the bulb of a sensitive thermometer was drilled in the upper bearing near its center, and this cavity was kept filled with oil in which the bulb of the thermometer was immersed, its protruding stem being protected from loss of heat by radiation by a non-conducting material. The indications of the thermometer are taken as the temperature of the journal. The weight was applied to the upper bearing at its center through a lever, the insistent weight given in the table (Tables II and III) is the lever weight increased to its equivalent direct weight in the vertical plane of the journals horizontal axis.

The quantity of oil used for lubrication was carefully weighed on delicate apothecaries scales and the temperatures of the journal and the surrounding air were noted every three minutes.

The apparatus was placed in an ordinary lathe which received its motion from the machine shop engine. This motion was transmitted to the journal by means of a Neer rotary dynamometer which directly measured the pull of the friction of the journal. The fraction which this pull in pounds is of the insistent weight on the journal in pounds, is the coefficient of friction.

If the product of the dynamometrical pull in pounds by the speed of the circumference of the journal in feet in a given time, be divided by the weight of lubricant consumed in the same time, an expression will be obtained for the comparative economic value of the respective lubricants."

Having defined the scope of lubrication and provided for sufficient grades of oil that will satisfactorily fulfill their needs, it is necessary that the operating personnel be possessed of sufficient technical information to enable them to select the proper item for each machine. To this end a table of uses has been prepared as a guide to the proper selection of an item. This table follows.

The Bureau of Supplies and Accounts purchases lubricating oil for the Navy on the above basis on an annual contract which pro-

Symbol 1075 Oil

Ice-machine cylinders.
Pneumatic tools.
Hydraulic jacks.
Hydraulic systems.
Refrigerating machinery (household type).
Quenching oil.

Symbol 2110 Oil

Main bearings of Parsons marine turbines.
High-speed steam engines.
Turbo-generators (100-kilowatt and smaller).
Motors and dynamos.
Line shafting.
Small miscellaneous machinery.
Quenching oil.

Symbol 2135 Oil

General machine oil.
Turbo-generators (larger than 100-kilowatt), using circulating oil system or water-cooled bearings.
Navy standard Norfolk motor-boat engines.
Straight line-shaft turbines.
Miscellaneous machinery with oil cups, electric motors, and dynamos.
Small ring-oiled bearings.
Small water-cooled gas and gasoline engines exposed to cold-weather conditions.

Symbol 2310 Oil

High-compression motor-boat gasoline engines.
Motor cars and trucks.
In place of mineral marine engine oil (wick feed.)
Power cylinders of small and medium high speed, 2 and 4 cycle Diesel and auxiliary compressors.
Coast Guard patrol-boat gasoline engines.
Stationary gas and oil engines.
Gasoline engines in small tractors and cultivators.

Symbol 3030 Oil

Aircraft engines.
High-speed gasoline engines.
Air-compressor air cylinders of Diesel compressors.
Single-stage air compressors delivering air at over 90 pounds pressure.
Ring-oiled bearings of small turbo-generators (without
Rotary pumps.
Passenger motor cars.

Symbol 2190 Oil

Main marine engines and turbines.
SO_2 ice machines (shipboard type).
Miscellaneous marine auxiliary machinery.
Vertical inclosed ice machines of the crank-case type.
Small 2 and 4 cycle Diesel engines.
Passenger motor cars.
Submarine Diesel engines and main motors.
Turbo-blowers.
Reduction-geared turbines.
Medium-size water-cooled gas and gasoline engines.
Farm lighting outfits operated under cold-weather conditions.
Horisontal air compressors and crank-case type air compressor s(except Diesel-engine air compressors).

Symbol 2250 Oil

Submarine Diesel engines and main motors.
Navy V type motor-boat engines.
Motor cars, trucks, and launches.
Small and medium size 2 and 4 cycle air-injection Diesel engines and auxiliary compressors.
Air compressors.
Stationary gas engines.
Speed reducers.
Farm lighting outfits.

Symbol 5190 Oil

For superheated-steam cylinder lubrication.
Swabbing rods.
Heavy deck and boiler-room equipment.
Additions to tunnel bearing grease.
Inclosed transmission and differential gears.

Symbol 6135 Oil

Counterflow stationary engines using dry or saturated steam at steam pressures of 90 to 175 pounds and operating on loads not less than 75 per cent of rated capacity.
Stationary engines using wet steam above 100 pounds pressure with loads less than 75 per cent of rated capacity.

Aircraft engines.
High-speed gasoline engines.
Air-compressor air cylinders of Diesel compressors.
Single-stage air compressors delivering air at over 90 pounds pressure.
Ring-oiled bearings of small turbo-generators (without cooling systems).

Symbol 3100 Oil

Aircraft engines.
High-speed gasoline engines.
Motor trucks.
Motor boats.
Motor cycles.
Large Diesels, 2 and 4 cycle, cylinders of.

Symbol 3120 Oil

Aircraft engines.
Power cylinders of large Diesels of 3,000 horsepower or more, except submarine engines.
Stuffing boxes of large Diesel engines.
Ring-oiled bearings of small turbo-generators without circulating system or water-cooled bearings.
Gasoline engines and Diesels in high-speed motor launches.
Tractors.

Symbol 4065 Oil

Reciprocating marine engines (wick feed).
For thrust bearings and bearings of auxiliary deck machinery.

Symbol 5065 Oil

Reciprocating marine engine and auxiliary, either hand oil, wick-feed, or drop oilers.
Extra heavy machine oil.
Shafting oil.

Symbol 5150 Oil

For dry saturated cylinder lubrication of land and marine reciprocating engines where condensate is utilized for boiler feed, ice making, etc.
Swabbing rods.
Heavy deck and boiler-room equipment.
Additions to tunnel bearing grease.
Inclosed transmission and differential gears.
Stuffing boxes of 2-cycle double-acting Diesels.

Steam at steam pressures of 90 to 175 pounds and operating on loads not less than 75 per cent of rated capacity.
Stationary engines using wet steam above 100 pounds pressure with loads less than 75 per cent of rated capacity.
Counterflow steam engines using high-pressure, high-temperature steam but operating at low loads.
Steam cylinders of noncondensing engines and pumps.
Small steam engines in public buildings, etc., operating on pressures from 60 to 175 pounds.
Steam shovels, locomotive cranes, steam hoists, reciprocating steam engines (inland waterways and reclamation work), control and manipulating steam cylinders, small uniflow engines operating on steam pressures under 160 pounds.
This oil can also be used on condensing engines where condensed exhaust is not utilized. If plant is equipped for proper separation, can be used on engines where condensed exhaust is utilized for ice making, etc.

Symbol 7105 Oil

Counterflow engines using wet steam pressure below 90 pounds, small steam engines operating on steam pressures under 90 pounds, low-pressure cylinders of compound steam engines where superheat is not used in the high-pressure cylinder.
Control and manipulating steam cylinders operating intermittently.
Small portable steam engines operating in field service.

Symbol 8135 Oil

Cylinders of very small compressors (except air injection compressors) in marine service used in connection with engines.
Model T Ford automobiles.

Symbol 8190 Oil

Cylinders of compressors in marine service used in connection with engines.

Symbol 9045 Oil

Electric transformers and switches.

vides for deliveries at all Navy Yards, Naval Stations and further at any railroad station in the United States. Because of this flexibility, 26 government offices other than the Navy supply their requirements from this contract. The widely varying types of uses with evident satisfaction testify to the soundness of the above method of treating the lubricating oil problem.

NOTES.

INDEX TO NOTES.

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By Dr. W. Wilke.

† Comparison of Operating Costs of the Engine Departments of U. S. Shipping Board Diesel and Steam Cargo Vessels.

By Captain R. D. Gatewood (CC). U. S. N., Retired, Member.

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* Paper presented at British Institute of Fuel, London, October 24, 1929.

† Read at the thirty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1929.

THE AMERICAN CONTRIBUTION TO THE USE OF PULVERIZED FUEL ON SHIPBOARD.

By Carl J. Jefferson, M.E.

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The title of this paper has been selected advisedly inasmuch as they who would do pioneer work must needs contribute largely from their store of physical and mental energy as well as from their financial resources. Furthermore in the case of American Shipping, the original incentive, created by comparatively high priced fuel oil, which instituted the development work towards the application of mechanical stoking of solid fuels has, during the last few years, been materially reduced. The prevailing prices of fuel oil as compared to those of coal in America, are such, that at present it is extremely difficult to substitute an equipment for the combustion of solid fuels in competition with liquid fuels which can effect a saving in the operating fuel costs of sufficient magnitude as to be attractive to the majority of the trade routes operating from our shores. It is quite probable that this is a temporary condition but it means that development work now being carried on must be done with a view towards the needs of the future rather than to obtain a solution of an immediate problem. It is therefore proper that our share in the development of pulverized fuel for use on shipboard be called America's contribution to an engineering art.

WHY AMERICA ATTACKED THE PROBLEM.

It must be confessed that the reasons for America attacking this problem were not entirely altruistic. In fact, the original incentive was produced by high price fuel oil with the prospect of a continued increase in this price.

In 1925 a group of engineers to whom had been intrusted the problem of effecting reduced operating fuel costs in the Government Fleet, owned by the United States Shipping Board, made a survey of the large power plants of the country to determine their policy in regard to fuels and methods of combustion. This group of engineers known as the Fuel Conservation Committee of the United States Shipping Board is made up of representatives appointed by the various technical societies of the United States, and consists of the following members:

Captain C. A. McAllister, President, American Bureau of Shipping.

Captain R. D. Gatewood, C. C., U. S. N., Manager, Maintenance and Repair Division, United States Shipping Board.

Colonel G. Bartlett, Representative of the Steamship Operators' Association.

Professor H. L. Seward, Yale University, Representative of the American Society of Mechanical Engineers.

Professor L. Chapman, Massachusetts Institute of Technology, Representative of the Naval Architects and Marine Engineers' Association.

Mr. F. Low, ex-President of the American Society of Mechanical Engineers, Representative of the Technical Press of the United States.

Mr. J. Fagan, Chief Engineer of the S. S. *Leviathan*, Representative of the Marine Engineers' Association.

Mr. F. Webster, of the Marine Congress.

Mr. W. Hayes, Operating Division, United States Shipping Board.

Mr. C. A. Greaser, Secretary of the Committee.

The writer of this paper has the honor of serving as the head of the section of the Shipping Board which promulgates and executes the policies instituted by the Committee.

The resources, advice, information and facilities of this Committee are not only used by the operators of Government tonnage, but also by the various private operators of the United States.

The survey of the various large power plants showed that practically all of them are using and are planning to continue the use of coal rather than oil, and there is a growing tendency towards the adoption of pulverized fuel as with it higher plant efficiency has been obtained, and furthermore, it permits the use of a number of low grade fuels which could not be successfully employed in any other manner.

The survey also showed it to be quite evident that marine use of pulverized fuel would require considerable development work, as the small furnaces of marine boilers are quite different from the furnaces used in large power plants. Furthermore the stresses and strains to which all marine machinery are subjected are not encountered in the usual shore plant and considerable modification of existing designs of pulverized fuel equipment would be necessary before it could be considered seagoing.

LABORATORY WORK PROGRAM.

It was decided by the Committee that the best way to attack this problem would be in the laboratory rather than by attempting an actual service installation.

The first essential in establishing the laboratory was to have unprejudiced, thoroughly competent, and enthusiastic personnel, as these are characteristics that must be applied to all research work.

To secure this type of personnel, application was made to and granted by the Bureau of Steam Engineering of the United States Navy for their cooperation in carrying on the desired tests. The shore laboratory work has, therefore, been conducted at the Fuel Oil Testing Plant, Philadelphia Navy Yard, under the joint supervision of the United States Navy and the Fuel Conservation Section of the United States Shipping Board.

The Scotch marine boiler chosen as the calorimeter for conducting these tests was selected because the great majority of the merchant fleet burning coal are equipped with Scotch boilers. Furthermore, the Scotch boiler with its limited water-cooled furnaces appeared to present the most difficult problem, and it was felt that the lessons learned on this boiler could be readily applied to water tube boilers. It has subsequently been found that the basic fundamentals established in the Scotch marine boiler tests do apply to the water tube boiler, but the actual application differs considerably, as the chilling effect of the water-cooled furnace of the Scotch boiler on combustion is quite different in its action from the reradiating effect of the refractory lined water tube boiler.

Sometime prior to the institution of the laboratory work at the Fuel Oil Testing Plant, in fact, in 1921, a series of tests attempting to use pulverized fuel were carried on in a Scotch marine boiler at Chester, Pennsylvania, under the joint supervision of the United States Shipping Board and the Bureau of Mines. At this time the conventional fantail burner, such as found in the large furnaces of shore plants, was used and no special attention was given to the degree of pulverization fineness. In fact, at that time neither the theories of combustion or reduction of material had advanced to a point where success was possible in attempting to apply pulverized fuel to the marine boiler.

Between 1921 and 1925, however, considerable experimental work has been extended in the use of pulverized fuel under boilers, and at the Sherman

Creek Plant Tests, conducted in New York, it was demonstrated that unusual results in limited furnace volume could be secured by means of high flame turbulence such as is obtained in the "Well Furnace."

The first series of tests carried on in the Navy Yard were purely of an experimental nature, the equipment which was installed being of such a design that it was impracticable for service conditions. It was, however, an application of the "Well Furnace" to each of the three furnaces of the boiler and it demonstrated that when the turbulence was maintained, pulverized fuel could be successfully burned in the furnace of a Scotch boiler. It was further demonstrated that combustion conditions were materially improved when the fuel was pulverized so as to have far greater amount of superfines than had ever been found necessary in shore plants.

The handicap of securing satisfactory distribution among the twelve burners which made up the three Well furnaces used in these tests was too severe to permit the use of this type of installation under service conditions.

While the Well furnace tests were being carried on, Admiral Dyson, U. S. N., and Mr. Peabody, were conducting a series of experiments with a burner that employed the Well furnace principle. Their burner, however, instead of having four nozzles, had an infinite number of nozzles, the coal discharged through an annular orifice and the radial streams impinged one upon the other, creating the turbulence necessary for rapid ignition and combustion.

This burner was installed for the second series of tests carried on at the Navy Yard, pulverization being accomplished by a moderate speed impact mill operated at approximately 1200 R.P.M. The merit of this type of burner was shortly established, but considerable difficulty was experienced with the type of pulverizer installed, as it would not maintain the degree of pulverization necessary for the various grades of coal used, and furthermore, a number of mechanical difficulties developed which made it desirable for the manufacturers to withdraw their mill from the test until it was further developed.

For the third series of tests the same burner was used in connection with a slow speed ball mill supplied by the Kennedy Van Saun Company. After a number of preliminary tests had been made it appeared that this equipment could be satisfactorily employed under service conditions, but before making an actual installation it was deemed advisable to simulate a trans-Atlantic voyage as nearly as possible in the laboratory. To this end a 240-hour test was carried on in order that it might be determined what effect ash deposit would have upon the performance of the boiler. Special mention is made of this test, for it is doubtful whether a test of such continuous duration will be necessary again, as it was then proved that pulverized fuel could be applied to the Scotch marine boiler for commercial use.

The photograph shown on Plate No. 1 is that of the personnel associated with this 240-hour test and includes a number of the men who have carried on the laboratory and sea service tests that have been conducted in adapting pulverized fuel to marine boilers in America.

While the third series of tests, terminating in the 240-hours test and leading to the first American seagoing installation of pulverized fuel, namely, the S. S. *Mercer,* were being carried on, a series of pulverized fuel tests were conducted by the Babcock & Wilcox Company at their Bayonne plant on a marine type Babcock & Wilcox Water Tube Boiler. The pulverizer used for those tests was the slow speed Fuller-Bonot Ball Mill. The burner was an adoption of the Lodi Oil Burner, so modified as to be used with pulverized fuel, the atomizer being replaced by a coal pipe.

This series of tests showed high efficiencies. Albeit the range of rates of combustion were not carried as high as those of the Scotch boiler tests,

they do, however, form a very valuable basis for future work which should be extended to cover higher rates of combustion, and to determine the effects of various types of furnaces and arrangement of tube banks and baffles. The investigation in relation to the adoption of pulverized fuel to water tube boilers is especially important, for the advent of higher steam pressures will necessarily force the use of water tube boilers in lieu of Scotch boilers, as the Scotch boiler, while having many admirable characteristics, is not suitable for pressures greatly in excess of 250 pounds.

The results of these tests have been given by Mr. Stillman in a paper presented before the Society of Naval Architects and Marine Engineers at their 1927 meeting.

Following the 240-hour test, the equipment used by Mr. Stillman at Bayonne was transferred to the Navy Yard and a series of tests using it were conducted on the Scotch boiler. The rates of evaporation were somewhat higher than those carried on at the Bayonne tests. The efficiencies obtained show, however, practically a continuation of the efficiency curve of the earlier tests.

Up until this point all of the equipment employed at the Testing Plant consisted of a single mill from which was distributed the pulverized fuel to the three furnaces and in all cases more or less difficulty in effecting satisfactory distribution of the fuel was experienced. In fact, distribution had shown itself to be one of the major problems associated with the adoption of pulverized fuel to a Scotch boiler. Static, mechanical and pneumatic distributors were experimented with and while fair results were obtained none of them could be considered entirely satisfactory.

The pulverizers used for all of the previous tests required a considerable amount of floor space, which necessitated structural changes in a vessel in order to convert it from either oil fired or hand fired coal to pulverized fuel installation.

The next series of tests at the Navy Yard were made with a view of attacking both of the above problems. The equipment for this purpose was furnished by the Todd Shipyards Corporation who had been experimenting in their own plant for about two years.

The Todd equipment consists of a small pulverizer per burner, attached directly to the furnace throat. This arrangement eliminates distribution trouble and gives an equipment that can be installed without any modifications to the ship's structure.

The burner, which was also developed by the Todd Engineering Corporation, is of the turbulent type employing an internal diffuser that is not subjected to the deteriorating effects of the furnace temperature. The secondary air supply is controlled by a register of the oil burning type.

The series of tests of this equipment were the basis for the pulverized fuel installation that has been made by the United States Shipping Board on the S. S. *West Alsek*.

The question of distribution and reduced floor space also prompted the next series of tests which, at the time of writing this paper, are being carried on at the Navy Yard. The equipment consists of a Fuller-Lehigh Vertical Table Mill and Lodi Burners. Distribution in this case is effected by a divided head in the Mill as shown on Figure No. 2. From this figure it will be noted that the Mill consists of a rotating table which supports three large iron balls weighing approximately 120 pounds each. Rotation of the table causes the balls to revolve and grind the coal which has been fed down upon the table. The primary and carrier air is blown into the mill rather than drawn out by suction. The coal and primary air passes through an internal classifier and out through the divided head to the furnace. Very satisfactory distribution has been obtained during the tests of this pulverizer.

Installation of this equipment has been made on the *Dwight F. Davis* and on two colliers, the *Berwindglen* and *Berwindvale*.

During the tests above cited supplementary tests have been carried on, such as various types of distributors including the Blo-Gum which employs the injector principle. Different types of conveying machinery were included in the set up of the equipment and numerous grades of fuel were burned.

The tests have all been conducted, with the exception of those at Bayonne and at Chester, under the joint supervision of the United States Navy, United States Shipping Board, and the manufacturers of the different apparatus.

A great portion of the coal consumed has been donated by mine operators, not only of the United States but abroad. The number of offers for donations of coal have at all times been sufficient to permit a wide selection of the particular type of coal desired for any test and cover a wide range of ash content, ash fusing temperature and percentage of volatile, as well as various degrees of friability.

It is not practicable in this paper to give details of each of these tests. They have, however, been presented in a number of technical papers and the index shown on Table No. 1 lists the information so that it may be readily obtained.

NEED OF FURTHER LABORATORY WORK.

The art of burning pulverized fuel in small furnaces is still very much in its infancy. Manufacturers have begun to produce satisfactory equipment which, however, will require further test and development before it can be considered satisfactory for service conditions.

All systems that we have experimented with up to the present time have been strictly of the unit type. Although we believe that the strictly bin system is not practical for marine use it is quite possible that other systems may be developed which for certain work may be preferable, as for instance a modified bin plant which for lack of a better name we shall call an accumulator system, in which there shall be a small storage of pulverized fuel used in conjunction with the unit mill.

Certain basic data are needed for the design of the marine pulverized fuel plant, which should be determined in the laboratory. Chief among these required facts is the establishment of reliable co-efficients to be employed in determining the carrying capacity of air at various velocities and temperatures, for various degrees of fineness of the fuel and for varying percentage of the moisture content of the fuel. Also the effect on these co-efficients produced by different sizes and shapes of ducts used for conveying purposes.

Experiments are required in line of disposal of ash. While this is not serious under ordinary steaming conditions, it is a nuisance when blowing tubes, and with some grades of coal it presents a real problem from the point of view of slag in the furnaces.

As previously stated, extended experimental work with the water tube boiler is especially desirous, for this is the type of boiler that will be used in a great majority of new installations.

All the experimental work that we have done up to date has been with the view of adopting pulverized fuel to existing types of boilers. While this has been the most practical line of attack, there is no doubt that a boiler can be developed for the use of pulverized fuel, which will combine the advantages of the Scotch water cooled furnaces with the high pressure possibilities of the water tube boilers, thereby eliminating the slaging problem of the water tube boiler, and at the same time permit the use of high pressure steam.

The "grindability" of the various coals of the world and the air conditions for their best combustion conditions need to be determined. At present with existing boilers, pulverizers and burning equipment, fuels containing 16 per cent to 34 per cent volatile can be satisfactorily handled. This range, however, should be extended, especially in the direction of lower volatile content so as to permit the use of residual fuel which it is quite probable will become more and more abundant as carbonization processes become more prevalent.

The metallurgical feature of pulverizing equipment also requires attention so as to reduce the maintenance changes to a minimum.

LABORATORY LESSONS.

Our laboratory work carried on to date has brought out certain points that should closely be observed.

(*a*) Any minor mechanical defects shown in the equipment must be given special attention as seagoing service conditions are bound to make these defects much more serious.

(*b*) The usual practice of good laboratory technic, which requires check work, must be carried on so as to establish facts and not a mere hypothesis.

(*c*) Actual capacity ranges for various grades of fuel must be determined so that the designer of service installations shall know what can actually be done with the equipment.

(*d*) The equipment should be tested out with fuels having at least 7 per cent moisture content as well as with dry fuels, inasmuch as bunker fuel in transit is frequently subjected to atmospheric conditions which produces this percentage of moisture. In fact it has been noted that for five times out of the eight bunkerings of the S. S. *Mercer* the moisture content has been slightly in excess of 7 per cent.

(*e*) Actual fuel and water measurements are required as carbon dioxide readings, stack temperature and smoke conditions may indicate high efficiency, while as a matter of fact a considerable amount of the combustibles, in the form of a light gray coke, may be passing up the stack unconsumed.

(*f*) In the laboratory, the accessibility of all wearing parts must be developed as this is most essential for service conditions to assure continuous operation with minimum of shutdown, otherwise reduced steam pressures such as are incident to hand cleaning of fires will result and one of the major benefits of the use of pulverized fuel will be lost.

(*g*) The personnel at the test plant at the present state of the art should include the engineer officers who are to take the equipment to sea, as it is necessary to develop the technic of pulverized fuel combustion in the same manner as it has been necessary to develop it for other types of power generation.

(*h*) The data obtained at the test plant should, in addition to the completed report, be briefed into a ready reference for the use of the personnel who shall operate the equipment under service conditions.

LABORATORY PROGRESS.

The progress which has been made in the development of the adaption of pulverized coal to the Scotch marine boiler is indicated by Figure No. 3. On this chart is shown the boiler efficiencies and evaporation rates that have been obtained for the various series of tests previously outlined. It will be noted that the initial test made in 1921 shows a maximum rate of evaporation of 3½ pounds per square foot of heating surface, and that the highest efficiency obtained was only 61.7 per cent, this being obtained at the very low rate of evaporation of 2.2 pounds per square foot of heating surface.

The Well furnace test produced a much more practical range of evaporation, it being from 4.4 pounds to approximately 9.4 pounds per square foot of heating surface. The maximum efficiency which was obtained at the 5.6 pounds rate, however, was only 68 per cent.

The 240-hour run was made at evaporation rates such as might be expected under various operating conditions in a sea-going installation, being from 5.35 pounds to 8 pounds per square foot of heating surface. The maximum efficiency rate in this case was obtained at the somewhat above normal evaporation rate of 6 pounds per square foot of heating surface and amounts to 77 per cent.

The individual mill tests were extended to cover port conditions and show a range from 3.75 pounds, which incidentally is the maximum rate obtained during the 1921 series, up to 7.42 with a maximum efficiency of 84.2 per cent at the 5.3 pounds per square foot of heating surface rate of evaporation, which is approximately the normal capacity demand on existing cargo vessels.

On this chart is also shown an efficiency curve for a hand-fired test carried on at Chester in 1921. These tests were conducted by the United States Shipping Board and Bureau of Mines, the firing being performed by expert firemen supplied by the Bureau of Mines.

In comparing the hand-fired test with pulverized fuel tests, the fact must be kept in mind that there is far greater probability of duplicating the mechanical results in service than there is of reproducing hand-fired tests, as the effect of personnel is of far greater importance in the one case than it is in the other.

It will be noted that the 1927 tests were the first of the pulverized fuel tests to top the hand-fired results, and that the 1928-1929 results not only exceeded the hand-fired results but also slightly exceeded oil fired furnaces, this being due probably to the lower hydrocarbon losses of solid fuels as compared to liquid fuels.

AMERICAN MARINE INSTALLATIONS.

The laboratory results must act as the basis for the design of any actual installation which is to be subjected to service conditions. There are, however, several factors that present individual problems in every installation which cannot be solved by the duplication of the laboratory set-up and require special engineering analysis. Chief amongst these are:

(*a*) Boiler design which includes the design of the gas passes, and the pre-heating of the air used in the pulverizers and for combustion.
(*b*) Fuel conveying equipment.
(*c*) Preparation of the fuel so as to secure uniform feed to the pulverizers.

Up to date, all American pulverized fuel marine installations have been applied either to existing boiler plants or to boiler plants using orthodox design of marine boilers. Therefore the situation has been one where the pulverized fuel must be made to fit the boiler, and not a boiler designed for the use of pulverized fuel. In the case of the Scotch marine boiler, the furnace condition is fixed to a very large degree and can only be modified by the use of such refractory as can be installed to speed up flame propagation without interfering with the radiant heat transfer that is so important in the Scotch marine boiler. Care must also be taken that the refractory does not interfere with the proper cleaning of the furnaces.

The water tube boiler permits a somewhat wider range in the modification of the furnace design, such as use of water or air cooled furnace walls and also a great variety of each of these types of walls.

Baffling in a Scotch boiler is limited to the use of retarders and to possible deflector plates in the smoke boxes for drawing down ash contained in the flue gas. In all the installations we have made up to date, the tubes have been retarded 100 per cent by the use of comparatively short spiral ribbon type retarders. Air heaters and superheaters may also be considered a type of baffling used in connection with the Scotch boiler, their major function, however, is to absorb the waste heat of the flue gases and put it to useful work.

The water tube boiler can be fitted with a great variety of types of baffling, care being taken in all cases, however, to eliminate soot pockets and to avoid the danger of bird-nesting of the ash between the tubes. In this case, the air heater and the super-heater also plays a very important part in the type of baffling that shall be adopted.

A general statement may be made that the pre-heating of the combustion and carrier air by waste heat gases is to be employed wherever possible. It has been found in our laboratory that 300 degree pre-heated air can be used most advantageously with practically all grades of fuel. Considerably higher pre-heated air can be used with a greater variety of fuels but it is quite possible that higher temperatures will cause coking troubles when used with the very high volatile coals.

The conveying of the coal from the bunkers to the pulverizers presents a different problem in every installation. The ideal method of course is the gravity system and should be employed in any new installation. This, however, is practically impossible when modifying a hand fired or oil burning job to a pulverized fuel installation. Numerous types of conveyors have been employed and have generally given unexpected satisfactory results. The various systems that have been used up to date are the Screw Feed, Bucket Conveyor, Bucket Elevator, Belt Conveyor, Drag Conveyor, and the Skip Hoist.

In connection with the conveying problem, the satisfactory results that have been obtained are due probably to the fact that the maximum service demand put upon this type of equipment in marine installations is generally less than the rated design of the standardized equipment that has been developed for shore work, so that by using the standardized equipment and making only such modifications as will make it applicable to the marine installations; *e.g.*, guide rails to guard against rolling and pitching, you have a very large factor of safety.

Proper preparation of the coal before feeding it to the pulverizers is most essential, as uniform combustion rates are directly dependent upon uniform feed rates. Slack coal is always preferable but as the supply of this size of coal at present is somewhat doubtful at the various seaports, crushers are advisable. Standardized crusher equipment is also entirely suitable for this service as it in most cases has the same characteristics as conveying machinery; namely, the small standardized sizes are generally somewhat larger than would be required for the maximum demand on board ship.

Coal driers are not required with the unit type mill. However, wet coal is not desirable and undried washed coal is not satisfactory, the entrained water content increases very considerably the power required for pulverization of the fuel, as well as tending to pack the coal when passing through the feeders, thereby temporarily blocking off the fuel supply to the mills.

A picture sometimes can tell more than a thousand words, therefore a collection of sketches of the various American marine installations of pulverized fuel has been incorporated as a part of this paper.

TOW-BOAT "ILLINOIS."

The first water-born pulverized fuel installation placed in actual operation in our country is that which was made on the tow-boat *Illinois* which is now in service on the Mississippi River, operated by the Inland Waterway Corporation.

The installation, as may be seen from Figure No. 4, does not vary greatly from that found in a number of small shore plants.

The *Illinois* is fitted with two (2) water tube boilers and has an abundant floor space for the equipment. The conveying machinery includes a drag for hauling the coal from the main bunker to the elevator where it is raised to a ready bin and discharged by gravity to the pulverizers. The pulverizers are slow speed Kennedy Mills, one for each boiler. There are two burners per boiler, they being of the fan-tail type with diffuser. The bifurcated leads to the two burners presenting but very little trouble in regard to distribution. Both the primary and secondary air are heated by means of hollow tile furnace walls. The baffling of the boilers at first gave considerable trouble, but as the boilers are fitted with economizers the baffling was reduced to a checker work above the tubes inasmuch as the heat lost through the high flue gas temperature could be recovered in a large measure in the economizer.

S. S. "MERCER."

The first vessel operated in off-shore trade and the first vessel to cross the Atlantic with pulverized fuel is the S. S. *Mercer*. The installation on this vessel is shown on Figure No. 5.

The S. S. *Mercer* is a 9700 deadweight ton vessel originally fitted out as an oil burner, and converted to pulverized fuel by the United States Shipping Board at the Maryland Drydock Company's Yard in Baltimore. An oil burning vessel was selected for this initial installation, because there was no precedent to act as a guide to determine whether or not the pulverized fuel equipment would function satisfactorily when subjected to the rolling and pitching bound to be encountered on a North Atlantic voyage; and as the vessel was to be put in a regular commercial trade route it was considered advisable to have the oil fuel in reserve in case the pulverized fuel installation should prove to be a failure.

Fortunately it was found in actual service that the pulverized fuel equipment functioned very satisfactorily and on the initial voyage from the United States to Rotterdam only 100 barrels of oil were consumed. This oil was used during the periods when repairs were being made to the pulverized fuel auxiliaries. The 100 barrels consumption could have been entirely eliminated inasmuch as at no time were both of the pulverizers out of commission, but by using same no delay was made in meeting the sailing schedule of the vessel.

The equipment installed on the S. S. *Mercer* is quite similar to that which was used on the 240-hour test. The pulverizers being the Kennedy Slow Speed Ball Mills and the burners being the Peabody turbulent inside mixed type. Two jaw type crushers were installed in order that run of mine coal might be used. The bunker fuel may be fed directly to the daily supply bins through the 1½-inch grizzlies installed in the upper deck of these bins. It may also be fed through a 3-inch grizzly to an all-around bucket type conveyor which elevates it to the crusher, thence by gravity to the daily supply bunker and through a variable speed plow type feeder to the pulverizers.

The original plan of the S. S. *Mercer* was to supply the pulverized fuel to all nine furnaces from either of the two mills. It was found, however, during the dock trials that this scheme was impractical as the friction load

through the pipe and distributors was too severe for the primary air fan. It was also found impractical to operate the two mills in parallel since it was impossible to synchronize the two primary air fans.

The dock trials did, however, show that it was possible to operate the power plant of the vessel at full power with two boilers. The original plan was therefore modified so as to operate the two outboard boilers with a mill per boiler and the first two voyages were made with this arrangement.

Subsequently an attempt was made to operate two boilers from one mill and one boiler from the other mill. This has been fairly satisfactory, but the best results have been obtained with the mill per boiler type of operation, and this scheme of firing is now being carried on.

The boiler uptakes all discharged into a common stack and it is found that ash will settle down into the center boiler when the two outboard boilers are in service and the center boiler is dead. To eliminate this trouble sufficient velocity of gas is maintained in the center boiler by running a single oil fire.

The furnaces were originally fitted, in the forward end, with refractory cylinders 2½ inches thick and 27 inches long. To this original installation has been added a refractory lining one brick thick and one brick long at the horse-collar. Installation of this additional refractory has been found quite beneficial by experiments conducted at the Test Plant.

The primary air used for sweeping the mill and also for carrier air, together with the secondary air, is preheated by a tubular Howden single pass waste heat air heater.

The S. S. *Mercer* at the time of writing this paper has made eight trans-Atlantic voyages and has demonstrated the feasibility and relibaility of pulverized fuel for marine purposes.

The results of the initial voyage of the S. S. *Mercer* were sufficiently satisfactory to warrant drawing up the plans for six additional conversions. However, at about this time there was a marked decline in the price of fuel oil and the United States Shipping Board secured a three years' fuel oil contract at a remarkably low price.

It was therefore felt that in lieu of expending the funds that would be necessary for carrying out the six conversions, that the funds be used for further development work so as to produce improved equipment which could be used for conversion purposes, wherein the problems of distribution, modification of ship's structure, range of combusion rate to take care of port, as well as sea duty, would be eliminated.

As a matter of interest, however, Figure No. 6 is included as it shows the modifications in the S. S. *Mercer* equipment which were considered advisable after having made a service test. This installation is quite similar to that made in Great Britain on the S. S. *Hororata*. It will be noted that the primary air fan instead of being placed in line with the mill is placed above it. This modification is made in order to reduce floor space and also to guard against the worn balls of mill charge working into the fans and thereby causing excessive fan blade deterioration. Small mills were recommended so as to reduce the power consumption of the pulverized fuel plant and to permit operation with pulverized fuel at the reduced port steam load. The air heater was divided into primary air and secondary air zones so that the velocity of the primary air would not be tied up with the velocity of the secondary air. The pulverizer and primary fans are located in the fireroom instead of in a separate pulverizer room, in order that the equipment would be under closer supervision by the operating personnel.

It was planned to either bunker sized coal or to crush it at time of bunkering and to have gravity feed to the mills, thereby eliminating all conveying equipment.

All horizontal runs of the pulverized fuel piping were eliminated so as to prevent deposit of pulverized fuel in the pipe lines.

TOW-BOAT "TAMIQUA."

The S. S. *Mercer* installation stimulated interest in marine use of pulverized fuel on the part of private companies, both the shipowners and the pulverized fuel equipment manufacturers.

The first marine pulverized installation in the States to be fostered entirely by private capital is that made on the tug-boat *Tamiqua* operated by the Reading Railroad Company of Pennsylvania in the coastal trade where it is used for the purpose of towing coal barges.

The *Tamiqua* installation shown in Figure No. 7 is a conversion from a hand fired to a pulverized fuel installation.

Each of the two furnace Scotch boilers are fitted with Areo pulverizers and Peabody burners. The Areo pulverizer is a moderate speed impact mill operated at approximately 1200 R.P.M. Neither the secondary nor primary air is preheated in this installation, which of course affects the boiler efficiency from the combustion end and also on account of the failure to make use of the heat in the waste gases. The secondary air for this installation is supplied by a blower which is belt driven from the turbine drive shaft to the pulverizer. The fan discharges into an air box surrounding the burner registers.

No crushers have been installed as it is the intent to use only slack coal. The coal is hoisted from the main bunker by means of a bucket hoist and discharged into the ready bunker from whence it goes by gravity through a variable speed disc feeder to the pulverizers.

The original installation employed a direct connected turbine drive to pulverizers. This, however, did not prove satisfactory as the small turbine operating at only 1200 R.P.M. had so high a water rate, that it was necessary to operate an independent feed pump in addition to the beam feed pump of the main engine, which pyramided the steam consumption and resulted in a reduced overall plant efficiency. The original turbines are now being replaced by high speed turbines which drive the pulverizers through a gear reduction. The water rate of the new unit being approximately 50 per cent of the initial turbine and the reduced steam demand of the pulverizing plant will permit shutting down the independent feed pump and operating with the beam pump alone. A very marked improvement in overall plant efficiency should result from this modification. While the boiler efficiency of this installation is fairly good, it could be increased considerably by the use of air heaters and it is quite probable that they will be installed eventually.

TOW-BOAT "DWIGHT F. DAVIS."

The first boat to be built with the intent to use pulverized fuel only is the tow-boat *Dwight F. Davis*. This boat was built for the Inland Waterways Corporation at the Charles Ward Engineering Works, located at Charleston, West Virginia, and is now operated in the Warrior River Service, Alabama.

The *Dwight F. Davis* installation is shown in Figure No. 8. It consists of two Babcock & Wilcox water tube boilers with independent stacks. Each boiler being fitted with a large air heater. The air being further heated by acting as a cooling medium for the rear and front walls and the floors of the furnaces. Induced draft pulls the air through the air heaters around the furnaces through the registers into the furnaces where, as flue gas, it passes through the baffling and acts as the heating medium in the air heaters.

Each boiler is fitted with a Fuller-Lehigh Table Mill. Preheated primary air is drawn over from the double fronts and blown into the mills where it sweeps the mill and acts as the primary and carrier air for the pulverized fuel. A single Lodi Babcock & Wilcox burner is fitted to each boiler.

Coal is taken from the main bunker by means of a belt conveyor over a magnetic separator and elevated by a bucket conveyor which discharges into the ready bins, from thence by gravity through a variable speed disc feeder it is fed to the pulverizers. No crushers are employed as it is the intent to bunker entirely with slack coal.

S. S. "BERWINDVALE" AND S. S. "BERWINDGLEN."

The first seagoing vessels to be built in the United States in which it is planned to use pulverized fuel only, are the S. S. *Berwindvale* and S. S. *Berwindglen*. Both are colliers built for the Borwind-White Coal Mining Company by the Bethlehem Shipbuilding Corporation at their Fore River plant.

The installation of the S. S. *Berwindvale*, as shown in Figure No. 9, is in many ways similar to that made on the *Dwight F. Davis* with the exceptions of,

(*a*) A common stack for the two boilers.
(*b*) Two burners per boiler, distribution being effected by dividing mill head.
(*c*) Main bunkers are above mills and all conveying machinery eliminated.

The S. S. *Berwindglen* installation is similar to that of the S. S. *Berwindvale* except that two Scotch boilers are used instead of Babcock & Wilcox water tube boilers. The Scotch boilers have three furnaces each and distribution is accomplished by a three-way division at mill head as shown on Figure No. 2.

The S. S. *Berwindglen* employs forced draft while the S. S. *Berwindvale* uses induced draft.

It is contemplated to bunker both of these vessels with slack coal. However, in order to provide for bunkering with "run of mine" a large crusher has been installed above the bunkers on center line of vessel so that the coal can be crushed to 1½-inch maximum size when bunker coal is taken on board. Coaling hatches are located to port and starboard of crusher which will be used for the slack coal.

At the time of writing this paper neither of these vessels have been placed in commission but both of them will probably be in service before the paper is presented.

NOTE: Since the original writing of this paper the author has had the privilege of attending the trial trip of the S. S. *Berwindvale*. It is indeed most gratifying to see what an excellent installation can be obtained when building a vessel designed to burn pulverized coal, where full use has been made of gravity feed, simplified fireroom arrangement, high preheat of air, and convenient methods of ash removal.

S. S. "WEST ALSEK."

As a matter of personal interest it may be noted that this paper is being written on board the latest marine pulverized fuel installation made in the States during her initial voyage. I refer to the S. S. *West Alsek*, which vessel has been visited by a number of you during her port stays at Glasgow, Avonmouth and Cardiff.

The S. S. *West Alsek* is an 8800 deadweight ton cargo vessel originally fitted with hand fired Scotch marine boilers, and was selected by the United States Shipping Board for their second conversion to pulverized fuel, as she represents the typical cargo boat, built some ten or more years ago, with

which a great number of private shipowners are endeavoring to compete in their various trade routes, a good, seaworthy vessel but handicapped by the problem of hand firing a somewhat under-boilered job.

The equipment installed is a product produced by two years' development work on the part of the Todd Drydock Engineering and Repair Corporation, followed by six months' further tests at the Philadelphia Navy Yard.

It differs materially from all other plants that have been installed up to date, inasmuch as limited floor space and distribution problems have been overcome by the use of individual pulverizers for each burner. Figure No. 10 shows the installation on one of the three boilers and the conveying machinery for the port side, which is duplicated on the starboard side of fireroom.

The individual pulverizers which were built by the Erie City Iron Works for the Todd Company are in themselves different from previous pulverizer design, as they employ centrifugal force rather than impact or pressure crushing. The mills have two stages which, together with the fan that draws over the pulverized fuel and discharges it tangentially to the burners through flexible tubes, are mounted on a common shaft all rotating at approximately 3600 R.P.M.

Three pulverizers are joined together through flexible couplings and driven by a single direct connected turbine for each boiler. In effect this produces a single mill with three separate discharges. The feed for each pulverizer is under separate control so that any fire can be cut out of any one of the furnaces and still maintian fires in the other furnaces, which is a characteristic that is desirable during maneuvering conditions or when operating under reduced stand-by port load.

The S. S. *West Alsek*, being a conversion job, did not permit the use of gravity fed bunkers, but instead required conveying machinery in order that use could be made of the existing bunkers in the vessel. Coal is stored on this vessel in the shelter deck, the 'tween deck, engine room side bunkers, and also in a cross bunker which extends to the lower hold.

At the time of making installation on this vessel it was considered advisable to provide for the use of run of mine fuel. Therefore, crushers were installed and the conveying equipment was designed so as to bring the coal from the various bunkers to the crushers from which it was fed to the individual pulverizers.

Crushers have been installed on both the port and starboard sides at the 'tween deck level. Coal is fed to the crushers from the shelter deck by gravity, from the cross bunkers by elevators, and from the 'tween deck and engine room side bunkers by means of a chute leading to the downcoming side of elevators into the hopper at the bottom of elevator, from whence it is elevated to the crushers. The coal after passing through the crushers is fed by gravity to a screw conveyor which runs across the entire width of the fireroom. This screw conveyor may be operated either right or left hand, so as to take the discharge from either starboard or port crusher. The screw conveyor discharges by gravity into nine down-comers which in turn discharge into the ready bins above each of the nine pulverizers.

Both the primary and secondary air are preheated by means of a single pass Howden air heater. The primary air is given a second preheating by means of a unique arrangement. The burner diaphragms are hollow and shaped like a horseshoe. One leg of this horseshoe is opened to the Howden front from whence it draws preheated air, the other leg is piped up to the pulverizer. This arrangement serves the double purpose of cooling the diaphragm and giving additional preheat to the primary air.

The burner is fitted with an internal adjustable diffuser which gives the initial turbulence to the coal and the primary air. This turbulence is

augmented by the whirl motion given by the secondary air which enters the furnace through an adjustable register similar in design to that of an oil burning register. The internal burner diffuser has the advantage of not being subjected to the furnace temperature.

The furnaces have a refractory lining 27 inches long and 2½ inches thick at the forward end and 4½ inches thick by 9 inches long ring at the horse-collar. The boilers are fitted with draft gauges, pyromoters, CO_2 recorders and smoke indicators. They are also fitted with sight holes in the combustion chamber located concentric with the center line of furnace.

The stack has been divided into three sections, thereby supplying each boiler with an independent stack.

This installation, while subjected to a number of refinements, has on her initial voyage fully demonstrated the practicability of independent pulverized fuel feed for the individual furnace.

INSTALLATION LESSONS.

Each new actual installation has contributed its quota towards the knowledge necessary to develop a new art.

The operation of each of the installations in service has confirmed the basic principles that have been discovered in the laboratory which are essential for the adoption of pulverized fuel to marine boilers; namely, the need for very fine pulverization and high flame turbulence.

They have also emphasized the requirement of equal quantitative and qualitative distribution of the fuel between the various burners.

The problem of low cost pulverization has been met in installations up to date chiefly by means of economical auxiliary units. It is believed, however, that the economical use of exhaust steam from the auxiliary units, such as by the use of the Bauer-Wach System, would prove more advantageous.

The reliability of pulverized fuel for marine purposes has, by these various installations, been fully demonstrated.

The ability to use grades of fuel both as to size and quality more or less satisfactorily in a pulverized form, which fuels would be entirely unsatisfactory when fired in any other manner, has also been demonstrated.

The question of ash deposit, either on top side or in the furnaces, has been shown to be not at all serious on cargo boats. This phase of pulverized fuel operation will, however, require further consideration before pulverized fuel is applied to passenger vessels. It is, however, not a problem without solution. In fact, it is quite probable that modifications necessary to accomplish the desired results for passenger vessels will be comparatively simple and will be met by soot deposit pockets in the flue gas passages fitted with aspirators to be used in lieu of soot blowing up the stack.

The successful adoption of pulverized fuel to marine boilers is fairly recent, but its development has been remarkably rapid. In fact the only safe answer that can be made to the question as to "What is the latest design" is "It is on the trestle board."

THE ECONOMIC VALUE OF PULVERIZED FUEL.

Vessels are operated in order to carry cargo for their owners at a profit. It is therefore of prime importance to know to just what extent the adoption of pulverized fuel will affect operating costs.

The comparative performance of Diesels, hand fired coal and oil burners is pretty well known. Therefore if a comparative co-efficient between pulverized fuel and any other standard methods be drawn up it is possible to establish the position that pulverized fuel takes in the whole scheme.

TABLE NUMBER ONE

TEST	BOILER	PULVERIZER	BURNER	TECHNICAL PAPER	WHERE PUBLISHED	AUTHORS
CHESTER TESTS	SCOTCH	ARGO MEDIUM SPEED IMPACT	FAN TAIL	TESTS OF MARINE BOILER	UNITED STATES BUREAU OF MINES BULLETIN 214	M. KREISINGER J. BLIZZARD A. NUMFORD B. CROSS W. ARGYLE R. SHERMAN
SERIES #1 NAVY YARD	SCOTCH	FULLER STANDARD BALL MILL	WELL FURNACE	PULVERIZED COAL AS APPLIED TO SCOTCH MARINE BOILERS	VOL. 35 TRANSACTIONS AMERICAN SOCIETY NAVAL ARCHITECTS & MARINE ENGS.	COMMANDER J.S. EVANS U.S.N. C.J. JEFFERSON
				ADAPTION OF PULVERIZED FUEL TO MARINE BOILERS	INTERNATIONAL CONFERENCE ON BITUMINOUS COAL NOV. 1924-1928 VOL. I	COMMANDER J. BROSNEK U.S.N. C.J. JEFFERSON
SERIES #2 NAVY YARD	SCOTCH	SIMPLEX MEDIUM SPEED IMPACT	PEABODY	PULVERIZED COAL AS APPLIED TO SCOTCH MARINE BOILERS	VOL. 35 TRANSACTIONS AMERICAN SOC. N.A. & M.E.	COMMANDER J.S. EVANS U.S.N. C.J. JEFFERSON
				THE BURNING OF HYDROCARBONS UNDER MARINE BOILERS	VOL. 36 TRANSACTIONS AM. SOC. N.A. & M.E.	E.H. PEABODY
				MARINE DEVELOPMENT OF PULVERIZED COAL	JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS VOL. XL NO 3	COMMANDER J.S. EVANS U.S.N. R.C. BRIERLY
SERIES #3 NAVY YARD	SCOTCH	KENNEDY VAN SICKLE SLOW SPEED BALL MILL	PEABODY	PULVERIZED COAL AS APPLIED TO SCOTCH MARINE BOILERS	VOL. 35 TRANSACTIONS AM. SOC. N.A. & M.E.	COMMANDER R. JEVANS U.S.N. C.J. JEFFERSON
				ADAPTION OF PULVERIZED FUEL TO MARINE BOILERS	INT. CONF. ON BIT. COAL NOV. 1924-1928 VOL. I	COMMANDER J. BROSNEK U.S.N. C.J. JEFFERSON
BAYONNE TESTS	BABCOCK WATER TUBE	FULLER STANDARD BALL MILL	B&W LOOI	PULVERIZED COAL TESTS OF A MARINE WATER TUBE BOILER	VOL. 35 TRANSACTIONS AM. SOC. N.A. & M.E.	T.B. STILLMAN
SERIES #4 NAVY YARD	SCOTCH	FULLER STANDARD SLOW SPEED BALL MILL	B&W LOOI	DEVELOPMENT OF PULVERIZED FUEL FOR MARINE PURPOSES DURING 1927-1928	VOL. 36 TRANSACTIONS AM. SOC. N.A. & M.E.	COMMANDER J.S. EVANS U.S.N. COMMANDER J. BROSNEK U.S.N. C.J. JEFFERSON
				MARINE DEVELOPMENT OF PULVERIZED COAL	JOURNAL OF THE AMERICAN SOC. OF NAVAL ENGS. VOL. XL NO 3	COMMANDER J.S. EVANS U.S.N. R.C. BRIERLY
				ADAPTION OF PULVERIZED FUEL TO MARINE BOILERS	INT. CONF. ON BIT. COAL NOV. 1924-1928 VOL. I	COMMANDER J. BROSNEK U.S.N. C.J. JEFFERSON
SERIES #5 NAVY YARD	SCOTCH	TODD HIGH SPEED CENTRIFUGAL MILL	TODD	DEVELOPMENT OF PULVERIZED FUEL FOR MARINE PURPOSES DURING 1927-1928	VOL. 36 TRANSACTIONS AM. SOC. N.A. & M.E.	COMMANDER J.S. EVANS U.S.N. COMMANDER J. BROSNEK U.S.N. C.J. JEFFERSON
				ADAPTION OF PULVERIZED FUEL TO MARINE BOILERS	INT. CONF. ON BIT. COAL NOV. 1924-1928 VOL. I	COMMANDER J. BROSNEK U.S.N. C.J. JEFFERSON
				THE TODD PULVERIZED COAL BURNER	JOURNAL OF THE AMERICAN SOC. OF NAVAL ENGS. VOL. XLI NO 2	R.C. BRIERLY

NOTES.

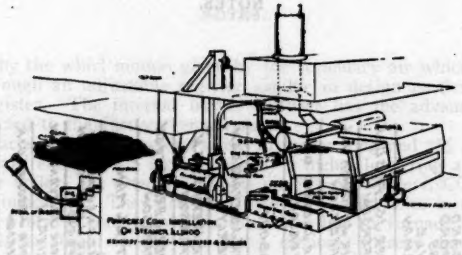


FIG. 4.

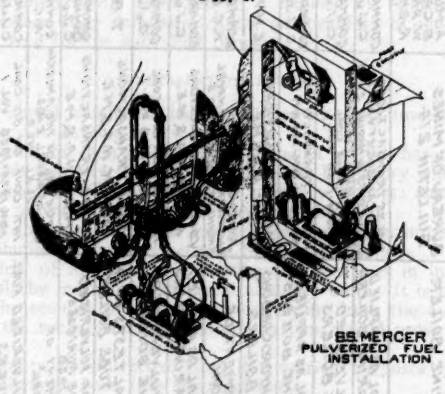


FIG. 5.

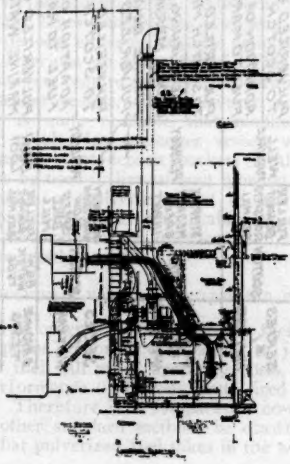


FIG. 6.

FIG. 1.

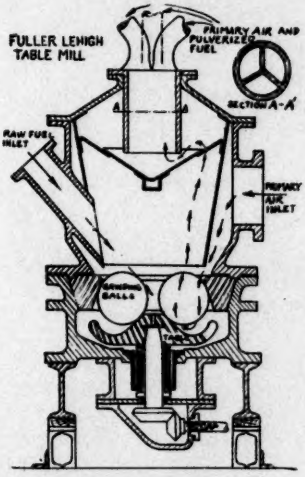


FIG. 2.

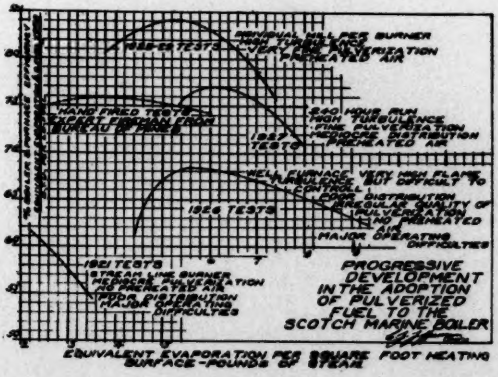


FIG. 3.

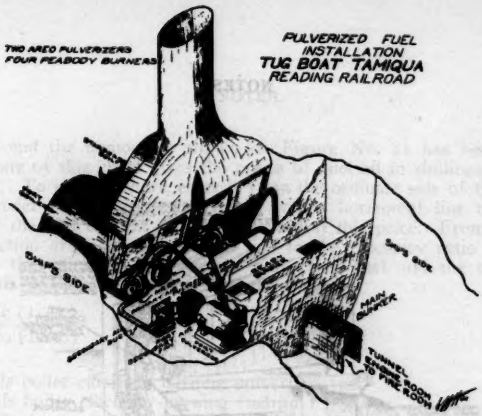


FIG. 7.

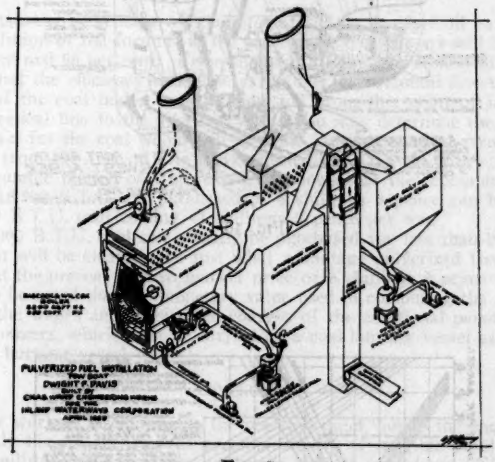


FIG. 8.

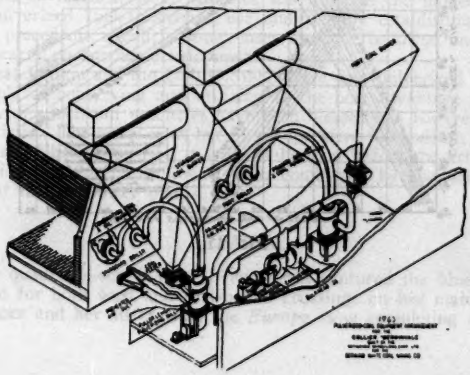


FIG. 9.

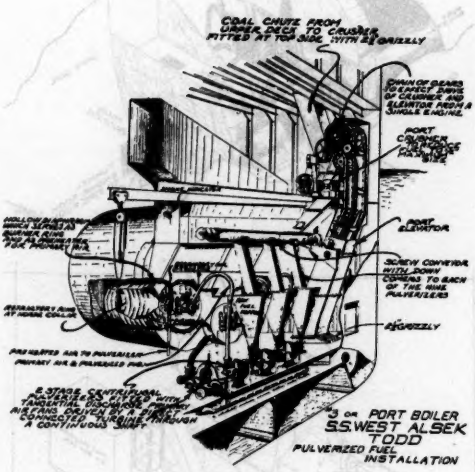

COAL CHUTE FROM UPPER DECK TO CRUSHER FITTED AT TOP SIDE WITH 2½" GRIZZLY
CHAIN OF BUCKETS TO ELEVATOR FROM A SINGLE ENGINE
PORT CRUSHER
PORT ELEVATOR
SCREW CONVEYOR WITH DOWN COMERS TO EACH OF THE NINE PULVERIZERS
2½" GRIZZLY
3 OR PORT BOILER
S.S. WEST ALSEK
TODD
PULVERIZED FUEL INSTALLATION

FIG. 10.

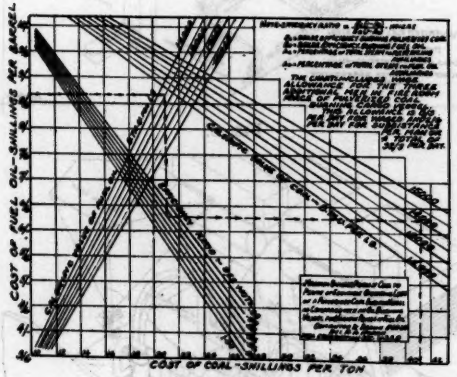

COST OF FUEL OIL—SHILLINGS PER BARREL
COST OF COAL—SHILLINGS PER TON

FIG. 11.

To this end the nomogram shown on Figure No. 11 has been charted. The ordinate of this chart gives the prices of fuel oil in shillings and pence per barrel. To use the chart, enter it from the ordinate side of the price of fuel oil under consideration and carry over a horizontal line to the heat content of the fuel oil that can be purchased at this price. From this point of intersection drop down a vertical line to the efficiency ratio line which represents the efficiency between the pulverized fuel and the oil burning plant. This efficiency is expressed by the formula,

$$\frac{Ec\ (1\text{-}Ac)}{Eo\ (1\text{-}Ao)}$$

in which,

Ec equals boiler efficiency burning pulverized fuel.

Eo equals boiler efficiency burning fuel oil.

Ac equals the percentage of total steam used by the pulverized fuel auxiliaries.

Ao equals the percentage of total steam used by the fuel oil auxiliaries.

The solution of the formula at the present stage of the art will be between 92 per cent and 96 per cent. From the point of the intersection of the vertical line and the efficiency ratio line carry over a horizontal line to the heat content of the coal under consideration and from this point of intersection drop a vertical line to the abscissa, where you may determine the price that can be paid for the coal which will give the same economical results as that obtained from the fuel oil originally selected.

The example plotted on Figure No. 11 shows that the economic performance of 18,500 B.T.U. oil purchased at 8 shillings 8 pence can be equalled by 14,000 B.T.U. coal costing 41 shillings 3 pence per ton.

As 14,000 B.T.U. coal can actually be purchased for less than 20 shillings per ton, it will be clearly seen just what advantage pulverized fuel has over fuel oil at the present British market price of 8 shillings 8 pence per barrel.

It is to be noted that the monetary value used in computing the monogram includes the wages and subsistence charges of the additional personnel, such as the trimmers, which must be carried on a coal burning vessel as compared to an oil burning vessel.

CONCLUSION.

Pioneer work is fascinating to the research mind, but to the engineer who must apply his efforts towards accomplishment of tangible results, it presents a multitude of difficulties due to the lack of precedents. The purpose of this paper is to show what America has done in the pioneer work of applying pulverized fuel to marine use and thereby establishing a certain number of precedents which in some cases must be followed and in others avoided to carry on further development.

Both Great Britain and the United States have sent pulverized fuel to sea. The principle of conserving and utilizing to the best advantage the natural resources with which our countries have been blessed has fostered the work of both nations. The art is new, but by co-operative methods on the part of the engineers of both countries, it will not be long before the art becomes a fixed practice that can be followed with confidence by the designers and operators of marine power plants.

BREMEN.

The new quadruple-screw liner *Bremen* has captured the blue riband of the Atlantic for both west and east-bound crossings on her maiden voyage. The new liner and her sister ship, the *Europa*, now completing at the yard

of Blohm & Voss, Hamburg, are undoubtedly the most outstanding mercantile vessels which have been placed in service since the war. Hitherto the Norddeutscher Lloyd have withheld details of the new vessels, but with the triumphant maiden voyage of the *Bremen* the ban of secrecy has to some extent been removed. Through the courtesy of the builders of the vessel, the Deutsche Schiff- und Maschinenbau A.-G., Bremen, we are able to give some hitherto unpublished particulars and illustrations of the machinery of the vessel. The principal particulars of the *Bremen* are as follows:

Length over all, feet and inches	938—3
Breadth, feet	102
Depth to main deck, feet	54
Gross tonnage, tons	49,864
Load displacement, about, tons	52,000
Corresponding draught, feet and inches	31—9
Block coefficient, about	0.60
Service speed, about, knots	28
Trial speed, about, knots	28½/29
Corresponding power, about, S.H.P.	120,000/130,000
Passenger accommodation:	
First class	800
Second class	300
Interchangeable first or second class	200
Tourist class	300
Third class	600
Crew	1050
Total complement	3250

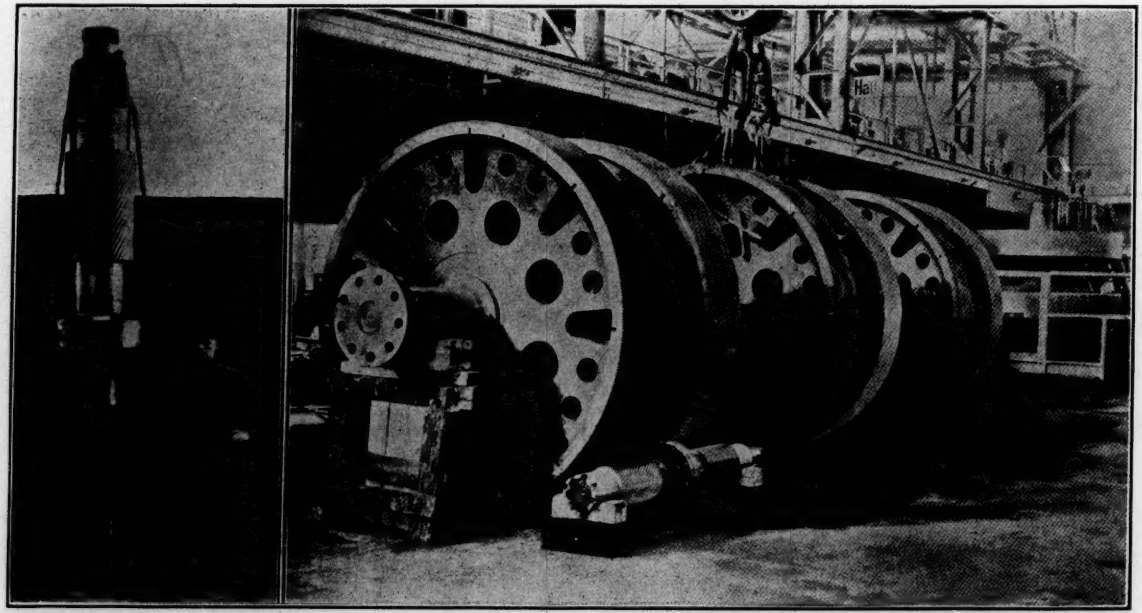
It is interesting to note that the launching weight of the vessel exceeded 32,000 tons.

The order for the *Bremen* was placed in December, 1926, the keel being laid by the Deutsche Schiff- und Maschinenbau A.-G. on June 17, 1927. She was launched on August 16, 1928, and commenced her maiden voyage on July 16, 1929, approximately two years from the date of the laying of the keel.

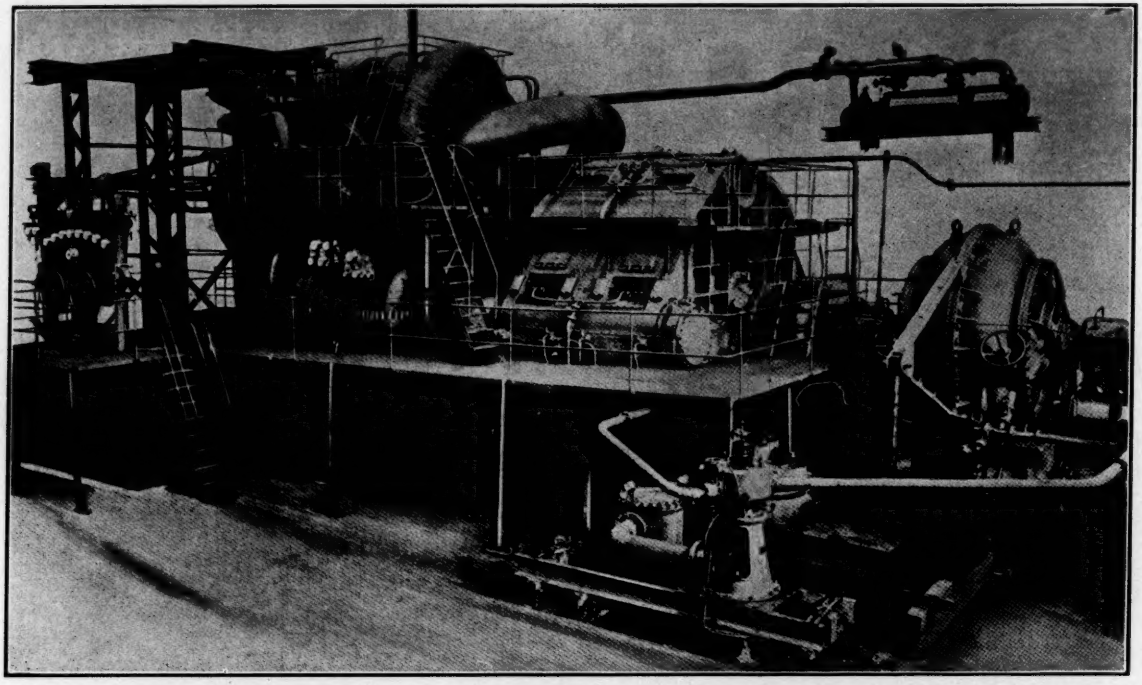
The *Bremen* is subdivided by means of fourteen watertight bulkheads, and it is claimed that if two adjoining compartments are open to the sea the vessel would still remain afloat. Furthermore, it is also claimed that if three compartments at the stern, or the first four compartments forward are completely flooded, neither the safety nor the propulsion of the vessel is seriously prejudiced. A feature of the vessel is the provision of very large unsinkable motor-propelled lifeboats, each boat holding about 145 persons. The lifeboats are so arranged that should the motor compartment be submerged the engine will still continue to operate. It is worthy of note that the boats are housed in Welin MacLachan davits. The Thermotank system of ventilation is used throughout the ship; an Oertz streamline rudder is provided.

PROPELLING MACHINERY.

The most interesting feature of the vessel is undoubtedly her propelling machinery, in view of the fact that the power with which she is provided is greatly in excess of that in any other mercantile vessel. In fact, the maximum power provided is about double that of the Cunarder *Mauretania*, in her original form.



PINION AND MAIN GEAR WHEELS FOR THE "BREMEN" AFTER LEAVING THE HOBBING MACHINES. NOTE THE DIVIDED PINION, THE DESIGN OF WHICH IS DISCUSSED IN THE ARTICLE.



ONE SET OF TURBINES AND SINGLE-REDUCTION GEARING FOR THE "BREMEN" ON THE TEST BED. THE TURBINE-DRIVEN FORCED LUBRICATION PUMP IS SEEN IN THE FOREGROUND, TOGETHER WITH THE OIL COOLERS AND STRAINERS.

The *Bremen* is propelled by four solid type four-bladed bronze propellers. It is worthy of note that the contract for the propellers was shared by two British and one German firms, J. Stone & Company, Limited, and the Manganese Bronze Company, each supplying one working propeller for the *Bremen* and one for the *Europa*. Each propeller is driven by a set of single-reduction geared turbines of the reaction type, these also having been manufactured and installed by the Deutsche Schiff- und Maschinenbau A.-G. Each set of turbines consists of high, intermediate and low-pressure ahead units with independent astern turbines. All turbine casings, including those for the low-pressure rotors, are of cast steel. The condensers are of the underslung type, and it is worthy of note that the condenser tubes are of the cupro-nickel variety supplied by Allen Everitt & Sons, Ltd. The main circulating pumps are electrically driven, each of the four pumps having a capacity of about 8000 tons per hour. The vacuum-producing arrangements are modelled on the most modern lines, the air-ejectors being of the four-stage type. Incidentally, in the illustration of one of the sets of turbines on the test bed which accompanies this article the air-ejector and its condenser, as well as the turbine-driven forced-lubrication pump, filter and cooler, are clearly shown.

The turbines run at 1800 R.P.M., this being reduced to a propeller-shaft speed of 180 R.P.M. The main gear-wheels for the *Bremen* are illustrated, these having been cut in the builders' shops at Bremen. The gearwheel bodies are of cast iron, the shrunk-on rims being of special steel; the pitch circle diameter is approximately 4000 millimeters.

The design of the main pinions is interesting and justifies description. The torque of the turbine is transmitted directly to the flange in the center of the pinion, and from there equally to the two tooth faces—an arrangement which minimizes the torsional deformation of the pinion and equalizes it as between the two halves, although it entails the adoption of a sleeve shaft or quill drive. The claws which are seen in one end of the L.P. pinion shown are for the reception of the turning wheel, the claw coupling for the turbine being fitted at the opposite end of the shaft.

Steam is supplied to the turbines by 20 oil-fired water-tube boilers. These boilers are of modified Thornycroft type, eleven being double-ended and nine of the single-ended type. The boilers are arranged in two widely separated boiler rooms, which arrangement explains the somewhat wide pitching of the funnels. Incidentally, the main machinery space is also divided into separate compartments, the auxiliary machinery being placed aft of the main engine room. The working steam pressure is 330 pounds per square inch, while the steam is superheated to a temperature of 700 degrees F. in superheaters which are integral with the boilers. A feature of the boilers is the provision of very large steam drums, these doubtless being provided with a view to maintaining a good head of steam when maneuvering. The boilers are oil-fired on the builders' system, and the closed stoke-hold system of forced draught is used. It is worthy of note that the boiler and steam-pipe insulation consists of Newall's 85 per cent magnesia coverings. The cold stores insulation also consists of Newall's Nonpareil cork used on the special Newsulate system. The insulation throughout the *Bremen* was carried out by Newall's Insulation Company, Limited, Washington, through their German agents.

For supplying electrical energy for driving steering gear, deck machinery, engine-room pump motors, ventilating fans and for the multitudinous domestic services on a large liner such as the *Bremen*, four 520 Kw. Diesel generator sets are installed in the auxiliary engineroom. These are six-cylinder trunk-piston four-stroke cycle airless-injection engines of the Weser-M.A.N. type. For emergency purposes two Diesel generator sets are provided, these being located on the boat deck.

Maiden Voyage—Bremerhaven–New York.

Departure from Bremerhaven, July 16, 6.18 min. (M.E.T.). Distance, 454 miles.

Departure from Southampton, July 17, 8 p.m. (M.E.T.); Distance, 82 miles.

Departure from Cherbourg, July 18, 2.12 a.m. (M.E.T.).

July.	Lat., North.	Long. West.	Daily Mileage.	Wind	Remarks (Weather, &c.).
	° ′				
18	49 49	8 59	288	S.W. 4–6	Cloudy, rather rough sea, patches of fog, reduced speed.
19	48 01	26 09	687	W.S.W. 5–3 6–7	Overcast, passing showers, rather rough sea.
20	43 30	41 37	704	N.N.W. 5–3	Fine and clear, moderate sea.
21	40 59	56 42	705	N.W. 2–4	Generally fair, slight sea, patches of fog, reduced speed.
22	40 30	72 22	713	N.E. 2–3	Overcast, slight sea, 1 p.m. mail-plane started, reduced speed.
		Rest	67		

Total distance, 3,164 miles; total time, 4 days 17 hours 42 min. average speed, 27·83 knots.

Maiden Voyage—New York–Bremerhaven.

July.	Lat., North.	Long., West.	Daily Mileage.	Wind.	Remarks (Weather, &c.)
	°	°			
27	40 13	68 28	249	S.W.–S.E. 2–3	2.42 a.m. (D.S.T.) passed Ambrose Channel Light vessel.
28	40 40	54 40	629	S.E. 2–5	Generally fair, slight sea.
29	43 17	41 11	641	W. 2–3	Fine weather, slight sea.
30	47 29	26 57	651	W.S.W. 4–7	Mostly overcast, rather rough to rough sea.
31	49 34	10 30	667	W. 7–8	Cloudy, rough sea and W. swell, 9.12 p.m. G.M.T. passed Eddystone Lighthouse.
		Rest	247		

Total distance, 3,084 miles; total time, 4 days 14 hours 30 min.; average speed, 27·91 knots.

MAIDEN VOYAGE PERFORMANCE.

In view of the wide publicity which has been given to the maiden voyage performance of the *Bremen*, this hitherto unpublished information is of more than ordinary interest. It will be noticed that on the outward run to New York a speed of 27.83 knots was averaged in weather which was certainly not conducive to the vessel giving of her best, while on the return passage the average speed was improved to 27.91 knots, the weather again being not too favorable.—"The Marine Engineer and Motorship Builder," September, 1929.

THE HIGHEST-POWERED MARINE DIESEL ENGINE.

ONE OF THE 10,000-HORSEPOWER UNITS FOR THE 27,000-TON WHITE STAR LINER "BRITANNIC."

The trials of the first of the two double-acting four-stroke Harland-B. and W. engines for installation in the 27,000-ton liner *Britannic* were carried out last month, and the illustration shows the engine erected in the builders' works at Belfast.

The trials were successful and the owners and their representatives commented upon the smooth and regular running of this unit, which is the highest powered unsupercharged marine Diesel engine yet constructed. It will develop an output of approximately 10,000 S.H.P., and it has ten cylinders 840 millimeters bore with a piston stroke of 1500 millimeters.

The general design of the engine is that of the normal Harland-B. and W. double-acting four-stroke type that has been fitted on so many passenger and cargo vessels. It will be noted that it has the standard cylinder bore and stroke which have been used with all of the more important engines of this class built since the first motors, namely, those which were installed in the *Gripsholm* and which had six cylinders. In that case the air compressors were driven by separate Diesel engines installed in an auxiliary engine room. The second engine is also ready for installation.—"The British Motorship," August, 1929.

MILWAUKEE.

NEW HAMBURG AMERIKA LINER WITH GEARED DOUBLE-ACTING DIESEL ENGINES.

The *Milwaukee*, which has just been built by Blohm & Voss, Hamburg, for the New York service of the Hamburg-Amerika Line, presents several novel mechanical features, having four Diesel engines geared to two propeller shafts. The ship will carry 456 cabin, 287 tourist-third, and 428 third-class passengers.

The principal dimensions are:

Length overall	175.07 m. (574 ft. 6 in.)
Length, on 8.84 m. (29 ft.) water-line	165.765 m. (544 ft.)
Beam	22 m. (72 ft. 2 in.)
Depth, moulded (to D deck)	11.2 m. (36 ft. 9 in.)
Draught	8.84 m. (29 ft.)
Deadweight capacity in above draught	10,320 tons
Sea speed with wind force not exceeding 3	16.5 knots.
Normal service speed	16 knots
Gross tonnage	16,700
Net tonnage	9612

The main machinery consists of four direct, reversible, double-acting two-stroke engines arranged in pairs and driving two propeller shafts through reduction gearing. The installation is the latest development of the Blohm & Voss system of marine drive. The object in view when this system was first adopted was the use of submarine engines in a merchant vessel where comparatively slow speed for the propeller shaft was essential. Extensive calculation and research were necessary to ensure that under all running conditions there was a positive pressure on the teeth of the gears, so that no tooth chatter could occur (see paper by Dr. Herm. Kahm, "Jahrbuch der Schiffbautechnischen Gesellschaft," 1924).

The above requirements were so satisfactorily met on the first two ships that were fitted with this type of drive (*Havelland* and *Münsterland*) that the Hamburg-Amerika Line, for whom they were built, ordered a passenger ship (the *Vogtland*) and a cargo ship (the *Friesland*), the former with the same drive as the *Havelland,* and the latter with two engines geared to a common shaft. As a further result of the satisfactory performance of these early ships, the Hamburg-South American Line placed orders for the *Monte Sarmiento* and the *Monte Olivia,* each with four oil engines driving two propellers through gearing. These ships have given satisfactory service for a number of years, and a third ship, the *Monte Cervantes,* has now been running for 18 months. All of the earlier ships were fitted with single-acting, four-stroke engines, but a further step forward with the geared drive is made in the present installation, where full advantage is taken of the small space occupied by, and the low weight of, the double-acting engine.

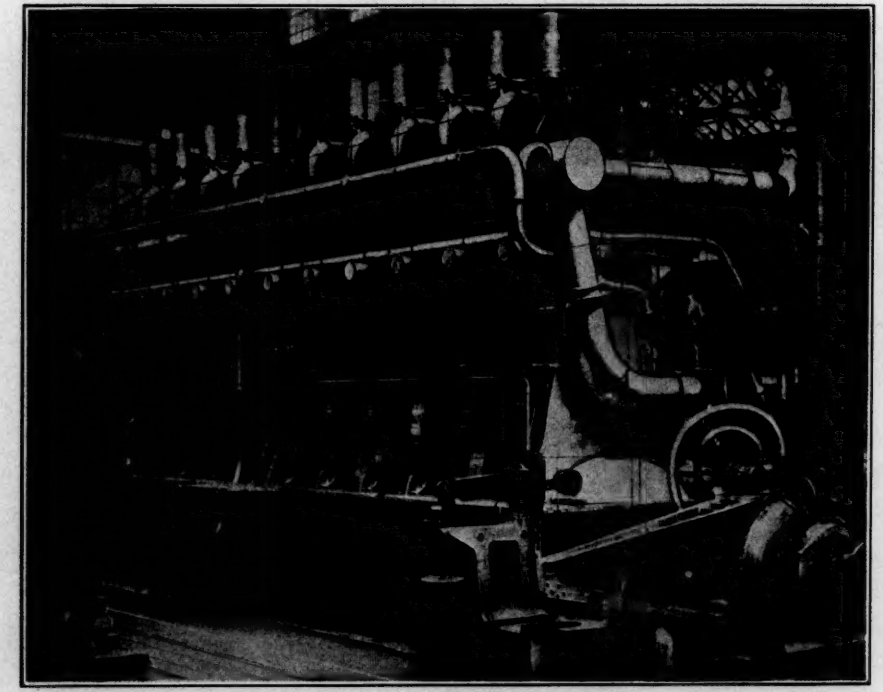
The main advantage of the indirect drive lies in the great saving in space and weight as compared with direct-coupled oil engines. The reduction in head room with fast-running engines is particularly valuable, as it makes the full length over the engine room through one or two of the deck spaces available for accommodation.

A further advantage claimed for the indirect drive on the *Milwaukee* is that the revolution speed of the engines is further removed from synchronism with the natural frequency of vibration of the ship's hull than would be the case if direct-coupled engines running at a lower speed were used. As a result of this, the probability of vibration is greatly reduced. Further, it must be remembered that a small, fast-running engine has less transverse force on the crossheads and smaller unbalanced forces and moments than a slow-running engine. Both of these facts assist in minimizing ship vibration. A further advantage of the system lies in the fact that, even if one engine should, for any reason, have to be stopped, the ship can still continue her voyage at three-quarter power. The small and light working parts can more easily be overhauled by the ship's personnel than could the heavy parts of direct-coupled engines.

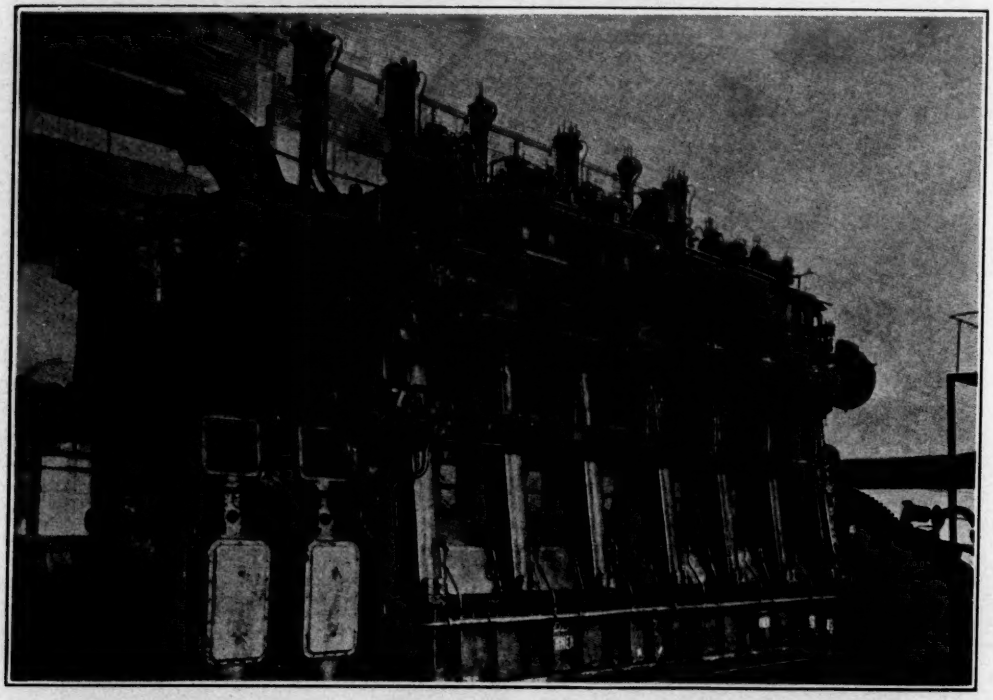
On the early ships on which each engine drove its own propeller, elastic shafts were fitted in order to reduce the critical speeds. On later ships in which two engines are coupled to each propeller shaft, these flexible shafts have been omitted as other means of avoiding torsional vibration have been found.

The crankshafts are direct coupled to the gear wheel couplings, and in this way the power is transmitted without loss between the engine and gearing. With direct coupling, the engines can be maneuvered as easily as a single engine, and the controls for both engines are arranged at a single control platform. The maneuvering of the engines is helped by stopping them quickly by means of automatic pneumatic brakes.

The main engines of the *Milwaukee* are of the M.A.N. six-cylinder, double-acting, two-stroke type built by Blohm & Voss. The cylinders are 500 millimeters (19.7 inches) diameter by 640 millimeters (25.2 inches)



ONE OF THE TWO 10,000 HORSEPOWER DOUBLE-ACTING HARLAND-B. AND W. ENGINES FOR THE "BRITANNIC" ON THE TEST BED.



ONE OF THE TWO-STROKE CYCLE DOUBLE-ACTING M.A.N. ENGINES OF THE GEARED MOTORSHIP "MILWAUKEE" ON THE TEST BED IN BLOHM & VOSS' SHOPS AT HAMBURG. THE ENGINE DEVELOPS 3000 H.P. AT A SPEED OF 220 R.P.M.

stroke and each engine develops 3000 H.P. at 220 R.P.M. The gearing reduces the propeller shaft speed to 110 R.P.M. The engines are fitted with air injection. A closed system of fresh-water cooling is used for pistons, cylinders and cylinder covers. Scavenge air is provided by three electrically driven turbo-blowers, each having a capacity of 800 cubic meters (28,400 cubic feet) per minute. Two blowers are sufficient to supply air for full-load running, the third acting as stand-by.

The exhaust gases of the main engine are passed through two waste-heat boilers of 200 square meters (2150 square feet) heating surface. The steam from these is used for heating, cooking, etc., The exhaust pipes are arranged so that the boilers can be by-passed, the exhaust gases going direct to the silencers, which are mounted in the forward funnel. Provision is made for oil firing in these boilers, so that steam is still available at times when the main engines are not running. Four electric generators are provided, driven by the latest type of six-cylinder, solid injection, four-stroke Blohm & Voss M.A.N. engines, having cylinders 400 millimeters (15¾ inches) diameter by 550 millimeters (21.6 inches) stroke, and running at 275 R.P.M. The pistons are not cooled, but the rest of the cooling is carried out by fresh water as in the case of the main engines.

AUXILIARY MACHINERY.

The auxiliary machinery in the main engine room includes the following:

2 fresh-water pumps for the main engine cooling system, each of capacity 820 to 1200 tons per hour.
2 salt-water pumps for circulating the fresh-water coolers, each of capacity 900 to 1200 tons per hour.
2 fresh-water coolers, each of 400 square meters (4300 square feet) cooling surface.
2 fresh-water drain tanks.
2 piston cooling water oil separators of the Blohm & Voss type.
1 make-up water pump of 5 tons per hour capacity.
3 gear-wheel type pumps for main engine lubrication system, each of capacity 90 tons per hour.
2 main engine lubricating oil separators, each of 90 square meters (970 square feet) cooling surface.
2 lubricating oil separators.
1 combined oil transfer pump and 200-ton ballast pump driven by a common motor.
2 daily-service tank fuel-oil pumps, each of 40 tons per hour capacity.
1 fire and sanitary pump of 75 tons per hour capacity.
1 bath pump of 20 tons per hour capacity.
2 drinking-water pumps, each of 20 tons per hour capacity.
2 sand filters for drinking water.
1 fresh-water circulating pump.
1 hydraulic water-tight door-operating installation.

The equipment in the auxiliary engine room includes the following:

2 starting air compressors, each of capacity 10 cubic meters (353 cubic feet) per minute.
1 emergency compressor of capacity 15 cubic meters (530 cubic feet) per hour.
1 combined fresh-water pump and auxiliary engine cooling pump for harbor service, both pumps having a capacity of 90 tons per hour, and driven by a common motor.
2 auxiliary engine lubricating oil pumps, each 25 tons per hour capacity.

2 auxiliary engine oil coolers, each 25 square meters (270 square feet) cooling surface.
4 main engine starting-air bottles.
4 auxiliary engine starting-air bottles.
2 boiler forced-draught fans.
2 Simplex boiler-feed pumps.
1 auxiliary condenser of 50 square meters (540 square feet) cooling surface.
1 bilge pump of capacity 75 tons per hour.
1 evaporator and distiller installation.

The ammonia refrigerator plant is installed in a separate compartment between the two propeller shafts. It consists of two compressors, each of 250,000 kilogram calories cooling capacity, together with their condensers, evaporators, moist-air cooler with ventilator, brine pump and cooling-water pump.

All auxiliaries are electrically driven, except the boiler-feed pump and the emergency compressor, which are coupled to a Diesel engine. The general arrangement of the machinery can be seen from the drawings of the engine and boiler rooms.

To provide the power required for the extensive electrical equipment, there are four 380-Kw. oil-engine-driven generators. Three generators carry the normal load at sea, the fourth being a stand-by. The generators are compound wound, and work in parallel with safety switch arrangements of the type developed by C. Meyer. Above the water-line a 25-Kw. emergency generator is provided. This is driven by an oil engine, and can be used for the most important services in case the main generating station has to be shut down. It is so arranged that it can be run in parallel with the main generators, if this should at any time be required. All electrical services on the ship are 220 volts, including the lighting installation, which has a total of about 5200 points.

The ship ran 24-hour trials on June 11. Six hours' running on overload was carried out with complete satisfaction, no trouble being experienced during this period. After the full-power trial there was a maneuvering trial lasting for about an hour and a half. All orders were readily answered, and it was particularly noticeable that the working cylinder relief valves did not blow on any occasion when the engine was started. The engine was reversed from full ahead in 25 seconds, the time being counted from the moment when the order was given until the engine reached 110 R.P.M. in the reverse direction.

The lowest propeller speed at which the engines ran was 30 R.P.M. The ship circled for a considerable time with the starboard engine running full ahead, and the port engine running full astern. The ship also ran for some time with the outer engines only driving the propellers, the inner engines being disconnected at the gear coupling.—"The Marine Engineer and Motorship Builder," August, 1929.

AN INTERESTING CORROSION PROBLEM.

Our readers will not fail, we think, to appreciate the novelty and interest of the facts and views expressed in the article, appearing elsewhere in this issue, on the use of a zinc-containing lubricant as a means of preventing the corrosion of ferrous metals. The features of the case may be briefly summarized. In a certain device subjected in use to rapidly repeated, local stresses corrosion of a ruinous order was found to develop quickly whether

the device was run dry or in the presence of an oil or grease lubricant. Prolonged investigation led the makers of the device to the belief that the corrosion was electrolytic in its origin, and that it ought to be possible, by mixing some substance with the lubricant, to convert the harmful corrosive effect of the electrolytic action into a beneficial protective effect. The substance eventually chosen was zinc oxide. When small quantities of this oxide were mixed with the lubricant little or no benefit was noticed. When, however, the amount added was some 50 per cent or more of the weight of the oil, it was found that all corrosion troubles disappeared and that the parts, instead of corroding, became coated with an extremely thin and apparently permanent film of metallic zinc. Examination at the National Physical Laboratory has definitely confirmed the opinion that the deposit is metallic zinc and has shown that the film has a thickness of 0.0004 millimeter or less, that is to say, less than the average wave length of white light, and therefore invisible beneath the microscope when viewed in section. The film is speedily established, and once established does not grow beyond a limiting thickness. That its thickness does not increase indefinitely is amply proved by the fact that the device shows no signs of jambing or seizing after it has been run for many hours in the presence of the zinc-depositing lubricant. These are the established facts of the case. They would seem to be of great scientific interest and to possess a wide significance from the practical point of view. Their theoretical interpretation must, however, for the present be approached cautiously. A considerable amount of experimental investigation has already been done on the subject, but much more, we believe, will be required before a generally acceptable theory of the action taking place can be formulated.

At an early stage of the studies which led to the discovery of the zinc-plating lubricant, the plating of the parts with cadmium in the ordinary way was tried as a means of inhibiting the corrosion. The film of cadmium speedily disappeared in use and left the parts open to the full corrosive action. An attempt was made to plate them with chromium, but it was found impossible to apply a film of this metal of sufficient thinness and evenness to avoid interference with the required dimensional accuracy of the device. Yet a zinc film of one to four ten-thousandths of a millimenter in thickness is found to give permanent protection to the parts. It would probably be found practicable to apply such a film by the ordinary processes of electro-plating, just as the cadmium film was applied. It is, however, almost impossible to believe that such a zinc film would be any more permanent and any more protective than the cadmium film. The permanency and constant thickness of the zinc film laid down by the zinc-containing lubricant cannot readily be explained otherwise than by asserting that the metallic zinc is being constantly deposited on the parts from the lubricant and that after the initial stage has been passed it is being oxidized off again back into the lubricant at an equal rate. After the parts have been run in the lubricant for many hours there is, we understand, no trace of metallic zinc in the lubricant. The zinc, it would therefore appear, passes through a closed cycle of de-oxidation and re-oxidation. What are the forces bringing about this chain of events? Are they purely mechanical or are they essentially electrical in their origin? The discoverers of the zinc lubricant believe, as we have said, that electrolytic action is at the basis of the phenomenon. They are probably correct, but a purely mechanical—or chemical—explanation should not at this stage be ruled out as unworthy of consideration. We know surprisingly little concerning the mechanical and chemical properties of the elements when spread out in extremely thin layers. Such knowledge as we do possess of them in that condition teaches us to expect properties which are not necessarily the same as, and which may, in fact, be greatly

different from, those exhibited by the same elements in ordinary masses. It is quite conceivable that the zinc oxide in the lubricant is de-oxidized by a purely chemical process, that the metallic zinc is rolled on to the parts in the manner in which silver was caused to adhere to copper in the manufacture of Sheffield plate, that the zinc is removed by ordinary abrasion as fast as it is applied, and that the cycle is completed by the re-oxidation of the finely divided zinc particles removed by the abrasive action. The electrolytic theory advanced by the discoverers of the zinc grease is, we feel, not at present—or, at any rate, not as at present revealed to us—wholly convincing. The dominating practical fact has, however, to be faced that under conditions, apparently reproducing those encountered in the actual device, differences of potential of as much as a tenth of a volt have been observed to exist between two steel surfaces, the stress between which was subject to a rapid cyclic variation. The presence of any such difference of potential cannot reasonably be ignored in the search for an explanation of the phenomenon observed. It is for this fundamental reason that we believe the investigators are right when they argue that the action taking place is of an electrolytic nature. Beyond that point we are not for the moment prepared to go. In particular we desire to preserve an open mind as to the origin of the electrical stress the existence of which has apparently been demonstrated. If, as is suggested, it is, in part at least, thermo-electric in its origin, it must be a much more complex manifestation of thermo-electricity than those to which attention has previously been directed. The zinc-plating effect is, we learn, shown definitely only when the contacting surfaces are both steel or iron. It is not exhibited if one or both of the surfaces is of brass or bronze. Thermo-electric phenomena have up to now been associated essentially with the contact of two different metals. The alternative, or supplementary, explanation of the origin of the potential difference, advanced by the investigators, appears to fit in with established knowledge more agreeably than does the thermo-electric theory. It is suggested that the principle at work may be the hitherto unsuspected converse of that in accordance with which the dimensions of a specimen of iron change when the metal is magnetized. This suggestion appears to include a hint as to why the plating effect is not observed in the case of non-ferrous—non-magnetic—metals.

The essential conditions which gave rise to the excessive corrosion encountered in the Keenok pinion, and which apparently are also required to exist before the zinc lubricant can exercise its protective effect, are, it seems, that the contacting surfaces should be of ferrous metal, that they should be subject to local stresses, and that the stresses should pass through a rapid cycle of variation. It is quite possible that the phenomenon has escaped general attention up to the present because practice, for a multitude of reasons, has unconsciously avoided the essential conditions required for its manifestation. It is equally possible that the phenomenon may provide the true explanation of certain mysterious cases of corrosion which are occasionally encountered. It is interesting in this connection to recall that in 1911 Messrs. Eden, Rose and Cunningham, in some experiments on the endurance of metals carried out at University College, London, found that some of their specimens after testing were rusted tightly into their holders in spite of precautions being taken to exclude moisture. They advanced the suggestion that the trouble was caused by a new phenomenon, the mechanical production of rust. In the course of some correspondence which ensued in our columns several other well-authenticated instances of apparently the same nature were disclosed, including the corrosion of ball bearings, keys and keyways, and two striking additional instances—one from the National Physical Laboratory—of the corrosion of test specimens. Conflicting opinions on the subject were expressed by several high authorities. It is today

interesting and significant to note the very complete manner in which the experience of the Keenok Company's investigators and the discovery of the protective action of the zinc lubricant support the views of those who held that the production of rust in the absence of moisture is a definite physical phenomenon and is not merely an idle fancy.—"The Engineer," September 27, 1929.

SURFACE HARDENING OF STEEL BY NITROGEN.

By H. W. McQuaid,* Detroit, Michigan.

A Discussion of the Position of Nitrided Steels as Engineering Materials, and the Practical Status of the Process.

The constant demands of the designing engineer for better materials of all kinds to produce mechanisms to meet the ever-increasing demands for stronger, faster, lighter, and safer equipment have resulted in a great effort on the part of the metallurgist to meet the needs of the engineer. The fact is that the requirements of the engineer for materials, particularly metals, to withstand greater stresses at higher temperatures, and to resist abrasion at higher speeds and higher unit pressures have created great problems in the metallurgical world which are being concentrated upon with all the available skill and resources of the American metallurgist. This has resulted in increasing greatly the importance of the metallurgist in modern industrial life.

Contributions which the metallurgists have made to engineering in the past decade have been quite noteworthy, and it is safe to say that the next ten years will show much greater developments than have the past ten years. One of the most recent contributions has been the developments in the surface hardening of steel by the addition of nitrogen instead of carbon to the outer portion. About 1925 the engineering world received the announcement that Dr. Fry of the Krupp Works in Germany had developed a process called "nitriding," whereby special steels could be treated with ammonia and thereby given a surface which was harder than any testing instrument could check, which would resist corrosion, and which would retain its properties at an elevated temperature. As is usual with such publicity, an immediate and widespread adoption of this process was predicted, but as is also generally the case, the subject developed slowly, and it was found that the special steels were high in price, hard to get, that the process required a very long time, and the results were uncertain. The metallurgists in investigating it found that it was the adoption of a much older process to a special steel and that the important contribution of Dr. Fry was not so much the process but the addition of aluminum to a steel to give it the properties claimed for after nitrogen hardening.

The engineer is interested not so much in the metallurgical details of a given process or a given material as he is in the new properties which can be developed by the process in a material so that the latter becomes to him more useful or more economical than the materials he is using. Up to the present it has been customary for the engineer, where surface hardness is required, to specify that the piece be case-hardened, and if exact size is desired that the piece, in addition to being case-hardened, be finish-ground. For many purposes this has been a very desirable method, especially since the low-carbon, easily machinable steels can be made into parts which have

* Metallurgist, The Timken-Detroit Axle Co.

Presented at a meeting of the Chicago Section of the A.S.M.E., Chicago, Ill., March 18, 1929.

a very high surface hardness, and which give very good results in such parts as gears, pins, bearings, etc., where a hard surface is advantageous. If special strength or special properties are desirable, the physical properties of the hardened surface can be improved by using special alloyed steels which are standard with the steel maker, and which are easily obtainable at a price which is strictly commercial. When high surface hardness is obtained with steels of high carbon content, such as tool steel, the depth of hardening is such that the piece becomes unsuitable for the purposes intended because of its tendency to be brittle, with consequent inability to withstand shock loading.

The case-hardening process, while it had the foregoing advantages, also had several disadvantages, some of the most important being the distortion encountered in the finished part, the tendency to soft spots, and the general lack of control due to the great variation which occurred with slight changes in the relatively high temperature employed. It is not unusual to have whole batches case-hardened too deep, to get a very high percentage of material which does not give the required hardness, or to have the warpage so great that the parts will not clean up in grinding. The disadvantages of case-hardening have not been so impressive to the engineer since they are difficulties for the metallurgist to handle, but nevertheless they add considerably to the cost of case-hardened parts.

In nitriding we are offered a material which can be given an extremely hard surface, which does not distort in the hardening process, and which is resistant to corrosion and to elevated temperatures. The object of this paper is to state as briefly as possible what the nitriding process is, what materials are required, what the disadvantages are, how it can be tested, and to what applications it is the most suitable. It is hoped that these will be presented so that they will satisfy the designing engineer's viewpoint.

STEELS FOR NITRIDING.

In June, 1913, Adolph Machlet, of the American Gas Furnace Company, was granted a patent on a process for giving iron or steel a superficial hardness by heating it in an atmosphere of ammonia at a temperature above 900 degrees F. As described by him this process had only a limited application due to the fact that while with ordinary iron and steel an extremely hard, non-corrosive surface can be obtained, the hardened zone is so extremely thin that it does not offer much resistance to abrasion and is of little value even under relatively low unit pressures. Where the unit pressures are very light, and the resistance to corrosion is an advantage, the Machlet treatment can be used to good effect. In 1924 Adolph Fry, of the Krupp Company in Germany, discovered that the addition of aluminum, chromium, and several other metals to the steel to be hardened resulted in a great increase in the depth of hardness obtained. He recommended that the process of hardening these special steels by ammonia should not be done at a temperature higher than 1075 degrees F., and in fact insisted that above this temperature the results obtained would not be commercially satisfactory. As described by him, the process required many hours to produce a satisfactory case, usually about 90 hours at 950 degrees F. It was also found that even at the lower temperature, the case of the nitrogen-hardened pieces was extremely brittle. To this process of surface hardening by means of ammonia the name of "nitriding" was given, which indicates that the hardening is due to the formation of metallic nitrides of iron, aluminum, chromium, etc.

Using the chrome-aluminum steels, which were quite difficult to make and machine, many articles were hardened satisfactorily, and the results obtained were sufficient to give the nitriding art considerable publicity. It was found,

however, that the addition of a small amount of molybdenum would materially reduce the notch brittleness of the nitrided piece, and at the present time practically all the nitriding steels contain molybdenum. Some of the special nitriding steels contain only molybdenum and aluminum as the characterizing elements. Others contain chromium and aluminum as these elements, with a small percentage of molybdenum added to increase the toughness.

The strength of the finished nitrided article is of course dependent upon two factors, namely, the strength and toughness of the hardened surface, and the strength and toughness of the core after the nitriding operation. If the hardened surface is extremely brittle, a load which results in fracture of the surface will be transferred to the core so quickly that fracture is immediately certain. It has been found with the nitrided material that, when the core has been properly treated and is large in proportion to the thickness of the nitrided case, the case will crack and the core will bend, a condition which is seldom found in the ordinary case-hardened piece. The core strength of the nitrided piece depends of course upon the alloy content and also upon the carbon content. Machining is affected very greatly by the carbon content, and hence for good machining it is desirable to have the carbon content low, but for high core strength and the best nitriding results it should be relatively high. For this reason nitriding steels vary in carbon content from approximately 0.30 to 0.50 per cent. Before nitriding, the special steels used will show when heat treated a yield point varying from 125,000 to 160,000 pounds per square inch, depending upon the carbon content and the nitriding temperature. After the nitriding is done at 1200 degrees F. the core strength is of course reduced. Inasmuch as the strength of the case really controls the strength of the parts and failure of the case results in failure of the part, the core strength is not of much value to the engineer. It has been the author's experience that very few engineers in designing parts to be made from case-hardened steel pay any attention to the special physical characteristics involved in such material; and blissfully make calculations based upon the estimated core strength, and neglect entirely the fact that failure of the case is practically always the criterion in failures of such parts.

Owing to the relative newness of the nitriding art, particularly as applied to the analysis of the material needed, the exact status of the special steels required is somewhat uncertain. Much work has been and is being done to develop improved steels for nitriding, and no doubt many suitable ones will be developed in the next few years. It is characteristic, however, of all the steels developed to date that the important element is aluminum, and the hardness imparted by nitrogen to aluminum-bearing steels is probably due to the great stability and hardness of the aluminum nitride. As mentioned earlier, molybdenum is of advantage in the nitriding process since it tends to improve the toughness of the finished article. Such elements as chromium, manganese, titanium, tungsten, vanadium, etc., form nitrides which tend to increase the hardness, although hardness readings as high as 1250 Brinell have been obtained with molybdenum-aluminum steels when the process has been altered to suit this particular type of steel. Apparently those elements which form carbides in the steel also form nitrides, with a resultant increase in hardness.

The cost of the steels employed in the nitriding process is higher than that of the steels used for case-hardening. This, to the engineer, is of greatest importance since it affects adversely the cost of the nitriding process in its competition with the case-hardening operation. At the present time one of the best-known and well-advertised nitriding steels is quoted at approximately 12 cents a pound for ordinary consumption, and as low as 7 cents for

requirements of 1000 tons per annum—this as compared with approximately 2 cents a pound for low-carbon case-hardening steels and approximately 4 cents for the better grade alloy case-hardening steel.

To overcome this handicap of initial high cost it is necessary that the results obtained in the finished article shall be ultimately economical as compared with the case-hardened steel part. There are many instances where the losses which are met with in case-hardening products due to soft spots, warpage, etc., are sufficient to make it well worth while to pay the higher cost of the nitriding steel with its lower losses in the finished article. In other cases the nitriding steels have to compete with the higher-priced stainless steels, and here the results are greatly in favor of the nitriding steels for such parts as pump shafts, valve seats, etc. There is no doubt but what the engineer will be specifying nitriding parts to replace stainless steel, monel metal, etc., in large quantities in the very near future. This is particularly true where not only is corrosion an important factor but hardness at elevated temperatures.

The nitriding process as now carried on consists in subjecting the parts to be nitrided to an atmosphere of ammonia at a temperature somewhere between 900 and 1200 degrees F. The time required depends upon the method and materials used, and varies greatly. It also depends to a great extent upon the use to which the nitrided article is to be put, varying from as little as one hour to as much as 100 hours.

It has been found that the best results are obtained when the nitriding steel is in what is known to the metallurgist as a sorbitic condition. This condition is obtained by quenching from above the critical range and then reheating and cooling slowly from a temperature below that range, generally in the neighborhood of 1100 degrees F. When the nitrided piece has been so treated the ductility of the finished article is greater and the tendency of the case to spall is greatly reduced. It is also important that the surface of the piece to be nitrided be clean, as some varieties of dirt and foreign matter exert a great influence upon the results obtained; and since it is not known exactly just what is harmful, it is best to prescribe a thorough cleaning either in a boiling caustic solution or gasoline. No surface on which satisfactory nitriding is to be obtained should be decarburized; *i.e.*, sufficient material should be machined off to get below any decarburization resulting from the oxidization met with in rolling or forging.

THE COMMERCIAL PROCESS OF NITRIDING.

The nitriding furnace may be of any design suitable to heat the retort employed to handle the work in process. However, the low temperatures lend themselves to electrically heated furnaces, and such furnaces have been found to be very desirable, especially when equipped with automatic temperature control. Gas-fired furnaces can be used just as well, and with equally good results, and the cheapness of such equipment might be a factor in deciding on which to build. The retorts in which the work is to be treated with ammonia should be made from a material which does not nitride easily, and it is therefore usual to construct them from an alloy high in nickel, of which latter there are many on the market such as nichrome, etc. Inasmuch as ammonia is expensive and its fumes are irritating, it is very desirable to seal the retort as tightly as possible and hence special covers are designed in which the seal is obtained by using various clamps, etc., in combination with gaskets of asbestos, nickel, and other materials. At the Timken-Detroit Axle Company satisfactory results are obtained by using a heavy nickel sheet for a cover and clamping this to the machined surface of the retort with a small amount of sodium silicate as a seal. In some furnaces the

cover or door is sealed with an oil or similar seal, and the work placed in baskets instead of retorts. This facilitates greatly the loading and unloading of the furnace.

In designing the nitriding furnace and laying out the nitriding department, one of the most important considerations is that of providing facilities for eliminating the ammonia fumes. Since these fumes are very irritating it is difficult to keep men working where they are pronounced. If the retort type of equipment is used, care must be taken to insure tight connections from the ammonia tank through to the water seal, and the escaping ammonia from the outlet pipe should be discharged in a small stream of running water. Special care should be given to handling the retort in and out of the furnace, and to closing up the latter when the retort is in place.

At the Timken-Detroit Axle Company the retort is provided with tracks on the bottom which run on nickel-chrome rollers. The retort is handled in the loading position in front of the furnace by means of these rollers. The inlet and outlet pipes for the ammonia are brought down close to the front of the furnace, and an asbestos mat provides the tight seal under the furnace door.

If the open type of furnace is used with a sealed cover, it is well to remember that the resistors operate at a much higher temperature than the furnace, and tend to increase the dissociation of the ammonia.

It should be borne in mind in designing a nitriding furnace that considerable time is lost if it is necessary to cool the work in the furnace; this is quite necessary in the basket type of furnace, but not in the retort type unless the high- and low-temperature method is employed in the nitriding operation. In this case much time will be lost in cooling the work so that it can be exposed to the air or so that the change from the high to the low temperature can be made, and it is well to provide a cooling system which will reduce the furnace temperature as rapidly as possible. In the ordinary nitriding operation a good cooling system will save from two to four hours per run, which is a very important consideration.

The standard commercial ammonia which can be obtained in most cities in 100-pound tanks has been found to be very satisfactory for the nitriding process, and the usual method is to pipe the ammonia from the ammonia tank through a reducing valve to the retort and from the retort to a water seal which indicates the pressure maintained in the retort. This pressure is ordinarily approximately an inch of water, so that rubber hose can be used conveniently to permit of movement of the retort without breaking the connections. It is well to equip the ammonia tank with a pressure-indicating gauge at the reducing valve, and to place the tank so that liquid ammonia will not enter the system.

It has been found that the nitriding process can be speeded up very greatly by increasing the circulation of the ammonia through the work. Apparently there is a tendency for the surface of the work to become passive, and for the nitriding action to proceed slowly unless fresh ammonia is continually brought into contact with the work. For this reason it is very desirable to circulate the ammonia as rapidly through the retort as possible, or to provide a circulating system in connection with the retort itself. This has been done by the Leeds & Northrup Company by adopting the Homo furnace for nitriding. This furnace contains a circulating fan which reverses its direction at regular intervals, and hence keeps a rapid flow and continuous agitation of the ammonia through the work.

When ammonia is heated, especially in contact with iron, it dissociates into hydrogen and nitrogen to an extent depending upon the temperature. It has been found desirable that this dissociation which takes place in the nitriding retort be kept to a certain minimum. Whether the important factor is the dissociation of the ammonia or the flow of the ammnoia is not

certain. It is the author's experience that the dissociation in itself can be disregarded and that the contact of the surface of the work with the ammonia, and therefore the percentage of undissociated ammonia in the retort and the circulation, is the important factor; *i.e.*, with a high degree of circulation the dissociation can be relatively great without any adverse affect on the work itself. Thus we have found in nitriding at 1200 degrees F., where the dissociation is hard to keep below 50 per cent, that very satisfactory results are obtained provided the flow of ammonia is sufficiently rapid through the work.

It is well to have a simple dissociation check apparatus set up to determine the percentage of undissociated ammonia coming from the retort.

After the type of furnace and the department layout are decided upon, the next important step is to determine at what temperature and on what cycles the nitriding is to be operated. According to Fry, no nitriding should be done at a temperature above 1075 degrees F. If this is done, according to the same authority the marginal layers will be brittle, tend to peel off, and give no surfaces suitable for parts which are subjected to severe strains. For this reason most of the nitriding which is being done is being kept below 1000 degrees F. with the result that the time required is often as much as 100 hours and the nitrided zones are relatively thin. They are, however, extremely hard.

This is not necessary, however, since it is possible to obtain a very hard, adherent case at temperatures as high as 1200 degrees F. which compares very favorably with those obtained at the lower temperature. In fact, some of the toughest and most satisfactory nitrided cases have been obtained by nitriding at a temperature of 1200 degrees. It has been the author's experience that as the temperature increases from 900 to 1250 degrees the surface hardness decreases, but the depth of the nitriding effect is greatly increased for a given time. Therefore, for many purposes a run of 20 hours at 1200 degrees, while it does not give a case which is as hard as would be obtained at the lower temperature, will give one which is very much deeper than would be had from a run, say, of 90 hours at 950 degrees. At the Timken-Detroit Axle Company the knowledge obtained from the depth-hardness checks has led to a cycle which has been found to be very satisfactory. This consists of a 10-hour period at 1200 degrees F., followed by a 15-hour period at 975 degrees. Using this method, Brinell hardnesses as high as 1200 have been obtained.

If the ammonia is mechanically circulated during the operation, the time cycle can be still further reduced, and very satisfactory results have been obtained using a 6-hour period at 1200 degrees F., followed by a 10-hour period at 975 degrees with a total cycle of approximately 18 hours.

If the main object of the nitriding is to insure a non-corrosive surface, such as on pump shafts and similar applications, the nitriding cycle can be reduced to a few hours; while if the maximum case is required it is believed that by using a cycle similar to the foregoing it is not necessary to nitride on a cycle totaling more than 30 hours. Since the proper nitriding temperature varies somewhat with the steels used, it is well to experiment in order to arrive at the most satisfactory one. The molybdenum-aluminum steels seem to require a higher temperature than the chromium-aluminum, and considerable experimenting with the various steels, cycles, etc., is advisable before setting the practice to any given cycle or steel.

If it is desirable to finish up with a surface which is as free as possible from discoloration, it will be necessary to permit the work to cool down in the atmosphere of ammonia to a temperature of not more than 350 degrees F. If the treated parts are exposed to the air at a temperature above this, they will have a blue oxide finish which is in itself softer than the nitrided

case beneath. For this reason, where the sealed furnace is used for nitriding, cooling means should be applied to the furnace to reduce the time required to cool to 350 degrees to a minimum.

RESULTS OBTAINED WITH NITRIDING PROCESS—APPLICATIONS.

If everything has gone well with the nitriding process and the above-mentioned precautions have been taken, the work when removed from the furnace or retort should possess a light silvery gray finish which is very smooth and absolutely file-hard. A dusty or powdery finish indicates that some detail has not been carefully attended to, either in the cleaning of the surface or the circulation of the ammonia. If the surface is very satisfactory but very thin and underlaid with a very soft zone, it is evident that the hydrogen has been more active than the nitrogen and that there has not been enough circulation of the ammonia over the surface to be nitrided.

The surface should not only be hard but it should be able to withstand heavy blows and considerable pounding without movement. The edges, if they possess even a very slight radius, should not flake when pounded with the hammer, and flat surfaces should withstand pounding with the peen end of a hammer without cracking. One of the best indications of good results in nitriding is a very smooth finish combined with an absolutely file-hard surface. If a section is ground and etched with a nitric acid solution, the nitrided case will be developed and its total depth can be ascertained easily. The depth indicated by the etch, however, is not indicative of the actual depth of file-hard material, as this is seldom more than 0.010 inch.

If the work has been carefully measured before nitriding, a recheck will show that warpage is commercially negligible although the nitriding process has been accompanied by a change in size. This increase in size will approximate 0.001 inch per inch of diameter, and has been found to be quite uniform, so that it can be allowed for in designing parts made from the nitriding steels. Very intricate parts have been made and nitrided because of the very desirable lack of change of shape exhibited by the finished material. Because of the high price of the nitriding steels and the newness of the art, the field of its application has as yet practically been undeveloped. It can be applied to any part which is to be subjected to heavy abrasive loads, and in fact wherever case-hardening is now used, with the possible exception of extremely heavily loaded carburized parts for which such steels as the 5 per cent nickels, nickel-molybdenum, etc., are now required. The nitriding steels are less resistant to high concentrated shock loads than are the high-alloy case-hardening steels, but in such parts as wristpins, timing gears, worms, journals, etc., the nitriding product is very superior to the case-hardened one. Most of the work done to date has been of an experimental nature, and thousands of tests are being made on all classes of work using nitrided parts. It is in everyday commercial production on parts such as pump shafts, vacuum-cleaner beater bars, valve seats to withstand high temperature and pressures, airplane-engine timing gears, and hundreds of parts requiring the highest degree of hardness with the least distortion. One consumer alone has need of approximately six tons of nitriding steel per week to satisfy his requirements.

Due to the price of the nitriding steels, many parts are excluded from nitriding because of the extra cost involved. Thus in the cheaper applications of the case-hardening process where plain carbon steels are used, the nitriding process cannot compete because of the initially higher cost of the material. For such parts as pump shafts and those where corrosion is an important factor as well as hardness, the nitrided part can be very economically employed since in this case it has only to compete with stainless steels which sell for a much higher price.

For such parts as bearings, gears, etc., whose manufacturers are at the present time probably the greatest users of case-hardened high-grade alloy steels, nitriding has still to be successfully demonstrated. There is no doubt, however, but what the lack of distortion and the superior hardness of nitrided parts will for these purposes some day be combined with toughness and machining factors which will make nitriding a standard method for treating such parts. Nitriding steels, due to their high alloy and carbon content, are much more difficult to machine than the low-carbon case-hardening steels. According to information which the author has, several very prominent European automobiles have used ring gears and pinions of nitrided chrome-aluminum steel, with extremely satisfactory results. In fact, if reports be true, one of the most expensive European models has practically eliminated case-hardening and replaced all parts formerly so treated with nitriding steels. Attempts so far to duplicate the European results in American gear applications have not been successful, although the author is convinced that this field will soon be open to nitriding steels.

From an engineering standpoint, serious consideration should be given to the use of nitriding steels as a means of combating corrosion and eliminating wear and distortion. The engineer should weigh carefully the advantages of the nitrided part as compared with the case-hardened part or the stainless-steel part.

As a word of caution, it must be admitted that some very poor results have been obtained with nitriding, but this has been because some simple factor in the process has been neglected. It is also well to remember that the nitriding art is in its infancy, and also that much effort and money are being expended at the present time to improve the process and develop cheaper and more satisfactory steels, and it is safe to say that the next few years will without doubt see the elimination of most of the objectionable factors at present entering into nitriding. All that is necessary is an insistent demand from the engineering profession that the metallurgist produce, and the results will follow to a degree depending upon the pressure applied.—"Mechanical Engineering," September, 1929.

LIQUEFACTION OF COAL.

ITS PURPOSE AND PRESENT STATUS.

By Dr. W. Wilke,* Director of German Dye Trust.

The possibility of exhausting the earth's crude oil supply must be reckoned with and means for covering our needs in liquid fuel from some other source should be developed. As base material for all such processes coal is used in most cases. The existing processes the author divided into the following categories:—distillation, cracking process, hydrogenation as well as coking and low temperature carbonization. Of these hydrogenation, or the direct saturation with hydrogen, has the best prospect of lasting success, because it gives the maximum amount of gasoline for instance. The following four methods of gasoline production are known:—(*a*) the so-called methanol-method of the German Dye Trust, (*b*) the catalytic low pressure hydrogenation of Fischer and Tropsch, (*c*) the catalytic high pressure hydrogenation of Bergius, and (*d*) the catalytic high pressure hydrogenation of the Dye Trust. Bergius' method gives out of 100 kilograms coal only 15

* Abstract of paper presented 8 Sept. 1929 at the Bremen meeting of Brennkraft-technischen Gesellschafft-Schiffbau 2 Oct. 1929, p. 462. Translated by E. C. Magdeburger.

kilograms gasoline and 6 kilograms lubricating oil, and for that reason has not received any large industrial application as yet, whereas the two methods of the Dye Trust have been already tried on a large scale.

The two methods of the German Dye Trust are characterized by the use of high pressures, high temperatures and catalyzators, which have the remarkable quality by their mere presence of choosing out of the number of possible reactions the one required by the process and in addition even accelerate these chemical changes. By suitable control of these means gasoline, oils and other hydro-carbons of any desired combination may be produced out of almost any substance containing carbon. The "Leuna-Werk" of the Trust is at present using brown coal out of the corporation's own mines (Weisenfels district) and is set to produce gasoline although the process may be at any time so regulated that fuel oils of different degree of distillation may be produced.

The apparatus required for the Dye Trust's hydrogenation processes is still in the stages of rapid development, hence the author did not wish to discuss details which may change very rapidly, but confined his remarks to essential fundamentals. There are four basic stages in the process, consisting of preparation, first hydrogenation (or so-called coal phase) in which coal paste and hydrogen, both under 200 at (3000 pounds per square inch) pressure, must be united, the second hydrogenation (or so-called gasoline phase) and further processing of gasoline.

The characteristics of the product are identical with those of a good quality gasoline. Successful operation requires in addition to the inherent qualities of the method itself also strict control, in which the measurement of the "intensity of knock" plays a very important role. As a measure of this "intensity of knock" serves the compression pressure at which the first signs of knock show up. A "knock scale" serves as a basis for comparison.

The installation at the Leuna-Werk even today, despite the fact that the process is in the stage of development, has a considerable output, and despite the increasing demands in gasoline its import can now be kept at a constant figure, because the output of the Leuna-Werk can cover the growth in the demand. The process is economically very important because it permits an almost complete conversion of solid coal into liquid fuel. The shipping industry is much interested in this process because it guarantees supply of synthetic liquid fuels, should natural oils become scarce as compared with existing conditions.

During the following discussion, Director Goos of Hapag mentioned that he heard a large installation using Bergius' process is in operation somewhere near Glasgow with a daily production of 600 tons of gasoline, corresponding to a daily consumption of 4000 tons of coal. The author replied that he had not heard anything about it, but Director Spilker said that he also received the same communication, without being able, however, to trace the product of this mysterious producer in any of the world markets. He believed that something entirely different, probably a shale distilling plant, gave rise to the rumor.

STANDARD OIL COMPANY APPLIES HYDROGENATION TO OIL.

Within the last few months have come indications of another radical change which may upset completely the entire motor-fuel business. As has been mentioned before in the press, the Standard Oil Company of New Jersey has entered into a contract with the German Dye Trust, known usually as the I. G. (initials of two German words meaning community of interests). The I. G. controls the Bergius hydrogenation processes, and it

was at first presumed that the Standard Oil Company of New Jersey was contemplating the far-distant possibility of the exhaustion of our oil resources and the use of coal as the source of motor gasoline as is now the case to some extent in Germany. It proved, however, that the Standard Oil Company of New Jersey was thinking of the present and not of the distant future, and that its intention was to apply hydrogenation not to coal but to oil, this taking the place of conventional cracking.

It has been stated officially that a unit to produce 5000 barrels per day is already under construction. No data are as yet available as to either the cost of the operation or the materials produced. It would appear, however, that Standard Oil would not have put up an expensive commercial unit unless these questions had been satisfactorily answered.—"Mechanical Engineering," Nov., 1929, p. 866.

COMPARISON OF OPERATING COSTS OF THE ENGINE DEPARTMENTS OF U. S. SHIPPING BOARD DIESEL AND STEAM CARGO VESSELS.

By Captain R. D. Gatewood (CC), U. S. N., Retired, Member.* †

A paper entitled "Performance of the Converted Motor Vessels of the U. S. Shipping Board" was presented before the Society at its meeting last year, and the discussion indicated a desire to be informed as to the costs of maintenance, which were not included in the paper. As a result this paper is presented with the tabulation of costs of operation of the engine departments only, of a number of vessels with three different types of propulsion, namely, Diesel engine direct drive, oil fired boilers with reciprocating engine drive, and oil fired boilers with turbine double-reduction gear drive.

A number of voyages are included of several vessels with the different types of drive, all operated by the same company and in the same service, namely, either from North Atlantic ports to India and return, or from North Atlantic ports to Australia and return by way of the Philippines and the Suez Canal; and since, for the most part, the various voyages were made during the same period of time it is reasonable to assume the vessels were all operating under generally similar cargo load conditions.

The tabulations which are presented have been computed from the actual charges against each vessel in the accounts of the U. S. Shipping Board, and, as previously stated, represent only charges against the engine department. There will probably be noted some seemingly inconsistent figures appearing in these tabulations, but these will be found due to the method adopted by the Accounting Division of the U. S. Shipping Board of requiring the voyage of any vessel to terminate at some particular time and date, and the next voyage to commence immediately upon the termination of the previous voyage. Consequently any charges incurred prior to a voyage termination are included in that particular voyage and any charges incurred after the termination included in the following voyage. The result is that not infrequently charges properly incurred in one voyage are carried over to the next voyage. This point is particularly applicable in regard to double-reduction geared-turbine vessels, where after a prolonged period in service it becomes necessary to renew reduction gears, and consequently the particular voyage in which this expense is incurred will carry a maintenance cost out of all proportion to the maintenance cost for other voyages. This, of course,

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† [Read at the thirty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1929.]

will be also applicable to Diesel-driven vessels where after a number of years service it will become necessary to renew cylinder liners, the consequence of which will be a high maintenance cost for that particular voyage.

Therefore, in preparing this paper, it was considered that the most valuable method of presenting the data to the Society was to present all the charges for the engine department carried against the several vessels and average the results obtained on both a cost per mile basis and a cost per day basis, giving also the average speed of the vessels.

Table 1 gives the results obtained with three vessels equipped with single-acting, 4-cycle McIntosh and Seymour engines.

Table 2 gives the results obtained with four vessels equipped with single-acting, 2-cycle Busch-Sulzer engines.

Table 3 gives the results obtained with two vessels equipped with double-acting, 2-cycle Worthington engines plus one vessel equipped with a double-acting, 2-cycle Hooven, Owens, Rentschler M.A.N. engine.

Table 4 gives the results obtained with two vessels equipped with 3 Scotch boilers, 210 pounds steam pressure, and a triple-expansion engine 2800 I.H.P.

Table 5 gives the results obtained with two vessels also equipped with 3 Scotch boilers, a triple-expansion engine 2800 I.H.P., one with 210 pounds steam pressure and one with 190 pounds.

Table 6 gives the results obtained with three vessels, one equipped with 3 Scotch boilers, 210 pounds steam pressure, and cross-compound 2800-S.H.P. Parsons turbines, double-reduction gears, one with 3 "B. and W." boilers, 200 pounds steam pressure, and cross-compound 3000-S.H.P. Westinghouse turbines and double-reduction gears, and one equipped with 3 Scotch boilers, 210 pounds steam pressure, single-expansion 2650-S.H.P. General Electric turbine and double-reduction gears.

Table 7 gives the average costs of the engine department per mile and per day for the ten Diesel engine propelled vessels, the four steam reciprocating propelled vessels, and the three steam turbine double-reduction geared vessels, together with the average speed, so they may be readily compared.

Table 8 gives the average cost of fuel oil per barrel used in the steam propelled vessel as compared with the average cost of fuel oil per barrel used in the Diesel propelled vessel.

In connection with Table 8, giving the average cost of fuel oil per barrel for steam and Diesel propelled vessel, it should be understood that in practically no instance in the voyages tabulated are the steam vessels able to bunker sufficient fuel oil for the round voyage, which consequently requires fuel oil to be bunkered at other ports than North Atlantic American ports, and all the oil bunkered is purchased upon the following specifications for boiler fuel:

Boiler Fuel.—Flash point not lower than 150 degrees F. Viscosity not greater than 300 seconds at 122 degrees F. Saybolt Furol. Water and Sediment: Sediment not over 0.25 per cent. Water plus sediment not more than 2 per cent.

Diesel Fuel.—Flash point not lower than 150 degrees F. Viscosity not greater than 100 seconds at 77 degrees F. Saybolt Furol. Sulphur not over 1.5 per cent. Water and Sediment: Sediment not over 0.25 per cent. Water plus sediment not more than 2 per cent.

It should be fully appreciated that the steam-propelled vessels are equipped with boilers and machinery designed and built during the war period and that no doubt much better fuel performance could be obtained with modern up-to-date machinery of this type and possibly in some instances lower maintenance costs, and/but it should also be realized that the Diesel engines were the first of either their size or type to be built and installed in ocean-going vessels in this country, so that it is to be logically expected many

Name of vessel	M. S. West Honaker, D.W.T. 8006				M. S. West Cusseta, D.W.T. 8006				M. S. Crown City, D.W.T. 8006			
Type of machinery	McIntosh & Seymour, 4-cycle single-acting, 2,700 H.P. Diesel engine				McIntosh & Seymour, 4-cycle, single-acting, 2,700 H.P. Diesel engine				McIntosh & Seymour, 4-cycle, single-acting, 2,700 H.P. Diesel engine			
Voyage number	1–52	2–53	3–54	4–55	1–51	2–52	3–53	4–54	1–51	2–52	3–53	4–54
Commencing	1–29–27	8– 7–27	3–10–28	9–19–28	1–17–27	5–15–27	11–30–27	5–12–28	2–22–27	8–23–27	2–15–28	8–11–28
Terminating	8–6–27	3– 9–28	9–18–28	3– 6–29	5–14–27	11–29–27	5–11–28	11–26–28	8–22–27	2–15–28	8– 1–28	3–19–29
Wages	9531.82	10761.68	9681.46	8399.88	5879.83	9928.30	8250.51	9929.55	9095.83	7405.36	8575.50	11204.88
Subsistence	1480.00	1360.00	1209.00	1073.00	881.91	1467.40	1102.00	1309 00	1166.00	1087 00	1272 00	1595 00
Fuel	18771.58	14731.21	16246.44	10966.96	15716.23	18720.07	14448.33	14635.09	21337 73	13567.43	15304 06	10670.04
Water	70.99	163.15	124.19	184.73	64.22	41.66	199 43	144.57	128 20	60 00	156 53	200.28
Consumable stores	920.25	2218.89	3032.96	3300.56	1116.25	3765.73	2062.94	2897.35	1497.65	2104.76	2417 21	1950.50
Laundry	67.30	81.50	107.00	112.80	24.50	84.70	40.30	74 00	86 60	93 50	114.50	111 00
Machinery spares	704.08	422.67	818.93	192.83	1396.66	1749.63	1131.48	1707.12	.....	334.27	122 16	334 90
Maintenance repairs	5750.48	3388.90	4169.21	12160.80	1006.00	215.00	778.50	.....	276.00	855.00	42.50	4705.12
Betterments	.....	.....	245.00	635.00	.....	.....	505.00	.....	.....	.....	.....	1189.00
Average cost fuel oil per bbl.	2.05	1.53	1.50	1.14	2.35	1.94	1.53	1.50	2.21	1.56	1.51	0.99
Total days	190	216	193	169	117.5	199	163	199	182	148.5	168.5	221
Days at sea	100.5	118	129.75	102.25	77.75	108.5	109.25	110.5	110.5	96	112.5	119.5
Days in port	89.5	98	63.25	66.75	39.75	90.5	53.75	88.5	71.5	52.5	56	101.5
Fuel bbls. at sea	8633	9123	10548	8364	6413.6	9314.7	9233.6	9346.4	9329.5	8449.1	9872	10278
Fuel bbls. in port	537.7	478	314	279	270.7	353.2	237 2	382	321	251.3	255.3	501
Total mileage	24777	28024	29448	25005	20748	27392	26280	28610	26254	23927	29479	29608
Expenses per 10,000 D.W.T. miles	1.88	1.48	1.51	1.85	1.87	1.64	1.36	1.34	1.60	1.33	1.19	1.35
Expenses per day	186.31				176.74				165.36			
Expenses per mile	1 33				1 18				1 09			
Observed running speed	10.51	10.35	9.97	10.58	11.44	10.89	10.31	11 10	10.72	11.14	11.34	10.88
Avg. obs. run. speed	10.35 knots				10.94 knots				11 02 knots			

TABLE 1.

Name of vessel	Sawokla, D.W.T. 9105				City of Rayville, D.W.T. 9050			
Type of machinery	Busch-Sulzer, 2-cycle, single-acting, 3,000 H.P. Diesel engine				Busch-Sulzer, 2-cycle, single-acting, 3,000 H.P. Diesel engine			
Voyage number	1–53	2–54	3–55	4–56	1–50	2–51	3–52	4–53
Commencing	7–17–27	11–20–27	5–25–28	11–17–28	4–28–27	12– 7–27	4–26–28	11–1 –28
Terminating	11–19–27	5–24–28	11–16–28	6–15–29	12–6 –27	4–26–28	11–1 –28	5–11–29
Wages	6198.04	9394.46	8831.71	10694.91	11126.77	6983.00	9901.19	9578.92
Subsistence	1050.24	1739.16	1477.32	1631.76	2403.70	1296 60	1865.88	1317.49
Fuel	16022.78	17315.55	12063.15	12223.64	19915.57	12620.80	15200.35	13284.02
Water	85.72	198.74	163.38	156.83	331.45	53.74	180.56	191.82
Consumable stores	2958.26	2976.39	3772.53	4292.32	4126.69	3247.38	2239.22	4218.90
Laundry	48.96	88.92	92.76	129 12	92.43	78.24	148.08	186.00
Machinery spares	87.22	511.50	1027 75	217 77	.....	1345.26	1061.40	186.99
Maintenance repairs	2583.58	6042.67	5906 03	7249.35	294 70	3219.94	1336.38	5776.37
Betterments	.....	575.00	66.00	.....	.....	940.00	515.00	378.00
Average cost fuel oil per bbl.	2.15	1.65	1.15	1.14	2.02	1.45	1.39	1.25
Total days	126	187	176	211	223	141	189.5	191 5
Days at sea	78.75	104.4	101.9	111.25	97.7	88	107	108.6
Days in port	47.25	82.6	74.1	99.75	125.3	53	82 5	82 9
Fuel—bbls. at sea	7223.5	10006	10151	10300	9216	8266	10472	10213
Fuel—bbls. in port	216.5	497	372	437	634	438	463	415
Total mileage	21149	29034	27618	28000	26082	23742	26760	28644
Expenses per 10,000 D.W.T. miles	1.51	1.47	1 33	1.44	1.62	1.39	1.34	1 35
Expenses per day	←		196.96	→	←		182 07	→
Expenses per mile	←		1 30	→	←		1 29	→
Observed running speed	11.95	12 15	11.52	11.44	11.36	11.57	10.76	11.67
Average observed running speed	←		11 77 knots	→	←		11 34 knots	→

TABLE 2.

Name of vessel	City of Dalhart, D.W.T. 9185				Yomachichi, D.W.T. 9235			
Type of machinery	Busch-Sulzer, 2-cycle, single-acting, 3,000 H.P. Diesel engine				Busch-Sulzer, 2-cycle, single-acting, 3,000 H.P. Diesel engine			
Voyage number	1–51	2–52	3–53	4–54	1–52	2–53	3–54	
Commencing	7–27–27	12–15–27	5–21–28	11–27–28	10–12–27	4–1 –28	10–20–28	
Terminating	12–14–27	5–20–28	11–27–28	6–25–29	3–31–28	10–19–28	5–1 –29	
Wages	6847.43	7956.38	9573.97	7573.70	8472.09	10651.03	9880.79	
Subsistence	1289.08	1417.39	1561.26	1354.56	1374.75	1552.20	1445.20	
Fuel	18107 11	13918.55	14781.01	11967.69	20697.51	19687.26	12572.59	
Water	36.71	88.90	127.29	68.22	39.22	84.78	94.60	
Consumable stores	3851.50	1960.50	2909.11	2843.35	2788.14	2522.79	3020.76	
Laundry	34.45	129.22	108.82	101.16	65.52	126.92	118.99	
Machinery spares		923.15	2288.24	337.33	627.12	874.68	89.29	
Maintenance repairs	94.00	2370.00	900.00	4587.10	599.83	2469.29	1025.09	
Betterments		780.00		1250.00	443.00	1955.00		
Average cost fuel oil per bbl.	2.03	1.45	1.45	1.25	1.87	1.60	1.19	
Total days	140.6	158	190.5	149.5	172	202	194	
Days at sea	91.8	91.7	108.5	90.2	100	118.1	113.4	
Days in port	48.8	66.3	82	59.3	72	83.9	80.6	
Fuel—bbls. at sea	8634	9222	9764	9266	10561.1	11842	10140	
Fuel—bbls. in port	284	377	429.8	349	482.4	474.5	426	
Total mileage	24843	26323	27570	24276	27660	31774	28870	
Expenses per 10,000 D.W.T. miles	1.33	1.22	1.27	1.35	1.37	1.36	1.06	
Expenses per day	←	19	1.26	→	←	18	1.83	→
Expenses per mile	←	1	19	→	←	1.17		→
Observed running speed	11.55	12.09	11.40	11.44	11.76	11.50	10.93	
Average observed running speed	←	11	62 knots	→	←	11.40	knots	→

TABLE 2 CONTINUED.

Name of vessel	Tampa, D.W.T. 9120			Unicoi, D.W.T. 9155			Seminole, D.W.T. 9132		
Type of machinery	Worthington, 2-cycle, double acting, 2900 H.P. Diesel engine			Worthington, 2-cycle, double acting, 2900 H.P. Diesel engine			Hooven, Owens, Rentschler, 2-cycle, double-acting 3680 H.P. Diesel engine		
Voyage number	1–57	2–58	3–59	1–53	2–54	3–55	1–54	2–55	3–56
Commencing	5–12–27	1–18–28	11–5 –28	3–25–27	11–29–27	7–7 –28	10–28–27	4–24–28	10–4 –28
Terminating	1–18–28	11–5 –28	5–8 –29	11–29–27	7–6 –28	1–14–29	4–23–28	10–3 –28	4–9 –29
Wages	12886.90	14424.91	9510.00	12450.47	11164.65	9606.30	9062.18	8130.66	9603.21
Subsistence	1705.67	1891.68	1749.24	2008.78	1717.17	1469.04	1352.16	1356.81	1357.20
Fuel	17733.43	16279.84	9410.25	18803.52	14013.09	15742.42	20389.76	12713.95	13706.01
Water	548.23	246.13	267.61	572.83	368.14	340.57	105.50	338.85	137.12
Consumable stores	3457.80	7208.36	2497.02	4082.90	3159.80	2779.58	2664.07	2744.97	3134.15
Laundry	108.10	140.88	111.72	125.84	89.31	94.32	61.44	56.81	78.24
Machinery spares	2349.22	2401.67	177.38	1028.71	674.92	1775.48	759.45	1605.65	116.14
Maintenance repairs	12515.16	10135.00	10867.10	12297.25	4228.94	6594.86	3573.32	1165.47	12041.48
Betterments	.....	8757.00	.....	.....	1265.00	.....	1113.00	.....	475.00
Average cost fuel oil per bbl.	1.66	1.37	1.09	1.94	1.67	1.45	1.51	1.15	1.04
Total days	251.5	292.5	184	249.5	220.5	192	179	163	188
Days at sea	113.8	115.8	79.25	112.75	86	123.2	109.4	90.6	105.9
Days in port	137.7	176.7	104.75	136.75	134.5	68.8	69.6	72.4	82.1
Fuel—bbls. at sea	10185.8	10686.4	8057	8959.3	7424.8	10495.9	13065.2	10720	12802
Fuel—bbls. in port	314.6	1212.5	612	720.1	972	350.8	424.8	330.5	428
Total mileage	26955	30069	21134	25870	21886	30779	30538	24615	27926
Expenses per 10,000 D.W.T. miles	2.09	2.24	1.80	2.17	1.83	1.36	1.40	1.25	1.59
Expenses per day	←		202.45 →	←		191.02 →	←		203.44 →
Expenses per mile	←		1.89 →	←		1.61 →	←		1.30 →
Observed running speed	10.31	11.42	11.27	9.75	11.05	10.77	12.02	12.08	11.39
Average observed running speed	←		11.00 knots →	←		10.52 knots →	←		11.83 knots →

TABLE 3.

NOTES.

Name of vessel	Editor, D.W.T., 9694				Arcturus, D.W.T. 9601				
Type of machinery	Reciprocating engine, 2800 I.H.P., 3 Scotch boilers, 210 lbs.				Reciprocating engine, 2800 I.H.P., 3 Scotch boilers, 210 lbs.				
Voyage number	1–77	1–78	2–79	3–80	5–64	6–65	7–66	8–67	9–68
Commencing	2–5 –27	1–3 –28	7–15–28	12–24–28	12–31–26	6–26–27	12–16–27	5–3 –28	2–8 –29
Terminating	8–10–27	7–14–28	12–23–28	6–12–29	6–9 –27	12–15–27	5–2 –28	2–7 –29	7–31–29
Wages	8097.56	8205.04	6901.81	7189.78	6668.69	7727.49	5934.09	11491.44	7454.06
Subsistence	1364.00	1460.52	1148.96	1191.26	1075.80	1305.24	992.04	1931.70	1264.32
Fuel	29414.36	31014.83	28718.03	24306.06	36656.16	34105.41	25071.20	35644.80	23619.44
Water	585.19	531.33	903.11	191.22	399.63	377.18	324.11	676.28	178.29
Consumable stores	1239.60	2280.08	1004.73	1008.93	1408.75	1539.22	1214.70	1266.84	963.12
Laundry	79.76	65.88	81.27	96.82	28.80	54.24	41.56	124.80	74.76
Machinery spares	.....	668.15	390.30	253.51	118.88	187.15	104.34	386.15	86.24
Maintenance repairs	5390.08	186.62	199.65	2389.90	2620.00	3028.71	1500.00	5525.00	1185.28
Betterments	.....	.....	.....	.....	.....	.....	50.00	.....	.....
Average cost fuel oil per bbl.	1.15	1.12	1.34	1.17	1.86	1.61	1.45	1.42	1.14
Total days	187	194	162	171	161	189	138	281	174
Days at sea	112.3	128.5	92.8	113.1	94.9	110.5	92.4	128	114.3
Days in port	74.7	65.5	69.2	57.9	66.1	78.5	45.6	153	59.7
Fuel—bbls. at sea	23199	24799	19084	18871	17207	19343	16105	21810	19347
Fuel—bbls. in port	2584.7	2952	2388.3	1986	2450.9	1794	1154	3361	1361
Total mileage	26598	28555	22148	24389	21994	26282	20993	28871	26004
Expenses per 10,000 D.W.T. miles	$1.79	$1.60	$1.83	$1.55	$2.32	$1.92	$1.75	$2.06	$1.39
Expenses per day	←	$233.27	→		←	→	$237.97	→	
Expenses per mile	←	1.64	→		←	→	$1.81	→	
Observed running speed	10.16	9.47	10.17	9.81	10.19	10.16	10.21	9.70	9.76
Average observed running speed	←	9.90 knots	→		←	→	10.00 knots	→	

TABLE 4.

Name of vessel	Egremont, D.W.T. 9976					Oak Park, D.W.T. 9455		
Type of machinery	Reciprocating engine, 2800 I.H.P., 3 Scotch boilers, 210 lbs.					Reciprocating engine, 2800 I.H.P. 3 Scotch boilers, 190 lbs.		
Voyage number	4–63	5–64	6–65	7–66	8–67	3–59	1–60	2–61
Commencing	10–24–26	4–23–27	10–20–27	4–8 –28	10–19–28	12–31–26	1–9 –28	6–27–28
Terminating	4–22–27	10–10–27	4–4 –28	10–18–28	3–25–29	6–4 –27	6–26–28	2–22–29
Wages	7780.70	7669.00	7305.28	8137.46	6655.05	6181.66	7250.54	10148.95
Subsistence	1246.44	1136.40	1270.44	1474.00	1244.01	1012.54	1275.12	1598.00
Fuel	42090.45	37385.41	33778 85	36706.28	24417.85	23505.13	38466.80	44440.29
Water	625.30	481.27	887.42	715.55	658.18	477.02	515.50	732.32
Consumable stores	1247.27	1231.12	1589.11	2540.10	1043.62	1516.15	1767.04	1545.52
Laundry	45.84	70.20	54.72	112.14	76.58	54.78	44.64	68.88
Machinery spares	275.40	114.80	164.12	703.67	178.26	1.09	222.21	18.64
Maintenance repairs	2844.95	4985.00	1358.00	491.26	5791.51	1673.62	3140.82	3873.99
Betterments					1271.00			
Average cost fuel oil per bbl.	1.87	1.83	1.65	1.44	1.14	1.16	1.66	1.48
Total days	181	180.5	170.5	194	158	138	170	241
Days at sea	99.1	93.4	93.2	120	101.6	103.9	121	144.1
Days in port	81.9	87.1	77.3	74	56.4	34.1	49	96.9
Fuel—bbls. at sea	20210	18452	18721	23531	19465	19024	21204	26907
Fuel—bbls. in port	2354	2003	1732	1991	1877	1227	1980	3209
Total mileage	22638	22544	21598	27868	22391	24506	28312	32011
Expenses per 10,000 D.W.T. miles	$2.49	$2.36	$2.15	$1.83	$1.85	$1.49	$1.97	$2.06
Expenses per day	←		$280.38		→	←	$272.37	→
Expenses per mile	←		$2.12		→	←	$1.76	→
Observed running speed	9.73	10.30	10.07	10.04	9.44	10.13	10.22	9.42
Average observed running speed	←		9.92 knots		→	←	9.92 knots	→

TABLE 5.

Name of vessel	S. S. Jalapa, D.W.T. 9407					
Type of machinery	3 Scotch boilers, 210 lb. S.P.–Parsons cross compound, double reduction gear, 2800 S.H.P. turbine					
Voyage number	5–62	6–63	7–64	8–65	9–66	10–67
Commencing	12–11–26	5–20–27	10–24–27	3–19–28	8–20–28	2–11–29
Terminating	5–19–27	10–23–27	3–18–28	8–19–28	2–10–29	8–2 –29
Wages	6875.51	6627.52	6384.45	6517.86	7494.84	7307.42
Subsistence	1351.80	1184.76	960.18	1202.52	1290.00	1153.33
Fuel	43769.06	40026.21	37050.24	27934.50	27199.12	22644.22
Water	405.98	646.60	683.20	659.96	721.33	630.59
Consumable stores	2380.39	1156.12	1187.90	1682.62	1140.04	951.58
Laundry	39.72	50.52	39.36	61.80	41.28	52.33
Machinery spares		751.40	343.16	260.79	109.85	649.53
Maintenance repairs	1708.87	3107.25	190.84	3067.36	13402.06	498.86
Betterments						
Average cost fuel oil per bbl.	1.88	2.00	1.75	1.52	1.35	1.19
Total days	160	157	448	154	175	173
Days at sea	104.6	93.5	99.1	97.3	98.3	100.1
Days in port	55 4	63.5	48.9	56.7	76.7	72.9
Fuel—bbls. at sea	20699	17729	18782	16426	17207	16574
Fuel—bbls. in port	2581	2322	2348.6	1985	3004	2453
Total mileage	23292	21445	21886	22027	32683	22474
Expenses per 10,000 D.W.T. miles	2.58	2.65	2.28	2.00	2.41	1.60
Expenses per day	←			29 3.27		→
Expenses per mile	←			2 12		→
Observed running speed	9.78	9.93	9.67	9.75	9.81	9.67
Average observed running speed	←			9 77 knots		→

Name of vessel	S. S. Algic, D.W.T. 8727					S. S. West Loquassuck	
Type of machinery	3 B. & W. boilers, 200 lb. S.P.–Westinghouse, 3000 S.H.P., Cross compound turbine					D.W.T. 8578, 3 Scotch boilers 210 S.P., General Electric 2650 S.H.P. Curtis geared turbine	
Voyage number	5–64	6–65	7–66	8–67	9–68	1–59	2–60
Commencing	1–15–27	7–13–27	1–12–28	6–2 –28	10–31–28	2–6 –28	9–7 –28
Terminating	7–12–27	1–12–28	6–1 –28	10–30–28	5–21–29	9–6 –28	2–14–29
Wages	8932.17	9204.00	7006.68	7461.28	8568.01	9182.80	6867.41
Subsistence	1290.33	1709.49	1246.23	1253.43	1622.61	1332.96	1229.40
Fuel	32723.50	41140.63	24296.03	22942.80	21084.30	27447.08	18304.02
Water	400.01	351.82	372.17	432.65	403.77	425.37	394.12
Consumable stores	995.40	1395.11	1323.41	1554.16	775.71	1669.04	1214.27
Laundry	47.94	49.38	43.31	40.00	68.17	75.24	44.76
Machinery spares		251.84	127.11	317.24	117.24	353.28	231.15
Maintenance repairs	3668.00	2535.80	2393.49	1967.98	4714.40	1102.96	1593.01
Betterments			20.00				1728.10
Av. cost fuel oil per bbl.	1.65	1.92	1.46	1.40	1.15	1.20	1.10
Total days	179	183.5	141.5	151	203	214	161
Days at sea	92.8	115.4	88.2	88.8	90.2	132.5	100.3
Days in port	86.2	68.1	53.3	62.2	112.8	81.5	60.7
Fuel—bbls. at sea	17594	19460	15134	14957	15690	20591	15187
Fuel—bbls. in port	2202.2	1998	1488	1456	2686	2332.5	1496.7
Total mileage	21859	26272	21715	21595	22130	27077	21854
Expenses per 10,000 D.W.T. miles	2.52	2.47	1.94	1.91	1.93	1.79	1.69
Expenses per day	←		250.41		→	← 195	19 →
Expenses per mile	←		1.89		→	← 1	50 →
Observed running speed	10.32	9.75	10.64	10 47	10.52	8.86	9.23
Average obs. run. speed	←		10.34	knots →		← 9	04 knots →

TABLE 6.

TABLE 7.

Average expenses engine department	Per day	Per mile	Average speed
			Knots
10 Diesel vessels	187.59	1.31	11.18
4 Reciprocating vessels	255.13	1.84	9.94
3 Turbine vessels	259.84	1.93	9.88

TABLE 8.

Average cost fuel oil per barrel	
Diesel vessels	1.52
Steam vessels	1.47

improvements could be made in future engines of this class. Indeed, in the later engines of the Diesel Program of the U. S. Shipping Board, which has just been completed, many improvements in design have been incorporated, and it is expected this will result in somewhat lower overall fuel consumption and much lower maintenance costs.

The conclusions which may be drawn from the tabulations presented here, which are as accurate as it is possible to compute them, are many and varied not only for the steam and Diesel engineer but also for the steam or Diesel motor vessel owner or operator. The outstanding conclusion is the difference in cost per day or per mile between Diesel and steam propulsion, in favor of Diesel propulsion, which difference is practically all obtained by the reduction in fuel consumption with fuel oil costing more money per unit and higher wages per day due to a fourth engineer being carried on Diesel vessels for training. Therefore, the objective of the steam propulsion engineer should be to endeavor to reduce the fuel consumption of steam propulsion units to a point comparable with Diesel propulsion, and the Diesel propulsion engineer should develop ways and means of utilizing the same fuel in Diesel engines which the steam engineer is now able to obtain and burn with success in boilers. The attaining of either of these objectives would be real progress from an engineering point of view and also from a shipowner's, if the oil companies do not then take all the cream off the milk.

There is also one further important objective for the Diesel engine manufacturer, and that is the reduction in first cost of Diesel power, so as to compare in cost with the equivalent in steam power.

BOOK REVIEWS.

"WHEN THE U BOATS CAME TO AMERICA." By WILLIAM BELL CLARK. PUBLISHED BY LITTLE, BROWN AND COMPANY, BOSTON, MASS.,.price $3.00. This is the story of the heroic performance and accomplishment in what to us was the big side show of the World War. The author accomplishes well his object of portraying the picture of the final attempt of the German Admiralty to win the war by carrying unrestricted submarine warfare to American shores.

In doing this no doubt he has acquainted many of our people for the first time with an accurate record of the many deeds of heroism and self-sacrifice which on account of propaganda and strict censorship during the War were never brought to light. From a comparison with official records, it is at once apparent that the author has arranged accurate statistics into a very readable chronological order of events which gives a true picture of the submarine depredations west of the 40th parallel and the measures taken by our own people to combat them from May, 1918, to the close of the War.

The author carefully restricts himself to his title text and by way of introduction outlines the various protective measures taken on our Atlantic seaboard.

In the next nineteen chapters are the chronological records of the visits of the various enemy submarines to our shores. Much of the text is in the form of official reports, extracts from diaries, and personal narratives of the various events. In this simple, unadorned way every chapter stands replete with the unswerving and determined will to win the War. It is the same story over and over of sudden and in many cases unseen attack and sinking of defenseless vessels, followed by the inevitable following hardships and privations of those who survived.

While here and there consideration was shown survivors, one cannot read many of these accounts without feeling that com-

pared with the ablity and esprit de corps of the early German Submarine Commanders the spirit of their submarine service had dropped to a low ebb. Just how much of this was due to orders they were acting under of destroying in many cases defenseless merchant vessels cannot be estimated.

The twentieth chapter, "The Epic of the Ticonderoga," is a detailed account of the lamentable and unfortunate loss of that vessel made particularly so from the fact that the *Galveston* unknowingly left the *Ticonderoga* to her fate.

The concluding chapter deals principally with the active countermeasures taken by our own submarines to protect our coastwise shipping and shores and the mopping up of mines following the Armistice.

One cannot read this book without reflecting on: The debt we owe Admiral Simms in FIRST changing the Department's policy of *playing safe at home;* SECOND, furnishing accurate, timely information of all "U" boat raids; the general apathy of our country to the possibilities of the submarine warfare striking home—even after the visit of the *U-53* and the *Deutschland;* the ability of the German submarines to carry on an aggressive campaign 3000 miles from any base; the great good sense of the American people in keeping their heads and not being stampeded into recalling patrol vessels from the war zone; the undaunted spirit of the personnel of our ships, both Naval and Merchant.

P. T. WRIGHT, COMMANDER, U. S. NAVY.

"SANDY HOOK TO 62°," By CHARLES EDWARD RUSSELL, PUBLISHED BY THE CENTURY COMPANY. PRICE, $3.50.

The Century Company offered more than $3.50 worth of entertainment and pleasure when they made available "Sandy Hook to 62°," by Charles Edward Russell.

As one closes this book at the end there is feeling of having lived with high adventure. There is a certain exhilaration of having associated with heroes, and Mr. Russell has so skilfully placed the surprisingly varied vicissitudes of the pilot's life, that one admires the calm mastery, and the do or die spirit, shown in the tremendous situations they have faced. At least it will be so

with every person who has a grain of salt in his veins, and with others—well, they will have a swelling of pride that they have such countrymen.

On the first tack Mr. Russell takes the reader through the circumstances attending the War of 1812, as the pilot boat and many pilots played a prominent part in that struggle. He punctuates it with incidents which should, and in many cases have, become classed among the great episodes of our history. Some of this material is not new; in fact, Mr. Russell refers to the Naval historians for his authority here and there. However, he has added a great deal of stirring stuff laboriously taken from old records and newspapers.

These chapters are finely set forth and it sets up a new train of thought regarding this war. It has been quite commonly believed to have resulted not altogether as we could have wished. Yet a point often overlooked is that we won in it what we never had before—the freedom of the seas. The vicious impressment of seamen passed away with the Treaty of Ghent.

As we had only twenty vessels of war all told against two hundred British on this coast and despite the fact that all our Navy covered itself with glory, through sheer force of numbers, it necessarily became a privateer's war. The results are imposing: There were 2300 merchant vessels captured with nearly 9000 guns. These vessels were valued with cargo at about $46,000,000.

These audacious privateers harassed the commerce of Great Britain from North Cape beyond the Arctic Circle to Cape Horn and in the China Seas. They even boldly cruised the Irish Seas and English Channel. The American seamen spurred to seemingly super-energy squared old scores in which six thousand of their countrymen had suffered involuntary servitude on British ships, and their exploits when they got the chance surpass the dreams of fictionists. They forced the insurance rates of England to over 30 per cent and so disrupted her commerce that American ships and seamen thereafter took a place of respect in the sun.

The reader of "Sandy Hook to 62°" is introduced to the three "Teasers" that gave the British commerce a lot of trouble; to Captain Wooster, a New York pilot, who when closely pursued by a British frigate threw his guns overboard and escaped. Then

"with his bare hands so to speak" captured an armed vessel (probably boarding with cutlasses), took her guns and continued his cruise. The reader learns to know other bold characters that make his blood tingle.

There follows a collection of enthralling stories of the pilot business—tremendous rescues, consummate seamanship, terrific exposure, heroic sacrifices, all with the splendid theme running through them, of self-effacement and devotion to duty as the tradition of the pilot service.

There are emotional incidents, such as the remarkable funeral of Capt. Freeborne, who had frozen to death on a stranded ship. She had just come out of the Tropics and her captain was freezing. Freeborne put his coat on him and put his sweater on the captain's daughter. The ship broke up and the bodies came ashore. As Capt. Freeborne was laid at rest on a hill overlooking the harbor a line of fourteen pilot schooners with flags at half mast, all vigorous commercial rivals, sailed by in tribute to a white and friendly soul.

And there was old Jacob Heath, left with two sailors and the cook on a battered, partially dismasted schooner after a collision in a ferocious blizzard. Those who were able had seized the chance and jumped for the rail of the steamer, and were safely brought to port, where they reported the loss of the pilot boat and Pilot Heath. Several days passed and the harbor front rubbed its eyes as it were at the sight of a schooner with bowsprit gone, only the foremast standing from which a single sail was set. Pilot Heath at headquarters read his obituary and he didn't like it. This was Friday afternoon, Saturday a new main-mast was stepped and he went to sea on Monday with a frost-bitten arm in a sling.

So, too, the chance must not be missed to know the redoubtable Boat-keeper Torgeson, a super-seaman. And there are many, many more.

Every locality has had its memorable storm with which all subsequent bad weather is compared as a standard. New York's measuring stick is the blizzard of March, 1888. Mr. Russell gives an interesting account of this unusual phenomenon, what it did to the city, and what it did to the pilot service. Imagine the historic tie-up in which no car moved in or out of that city for three

days, and on the sea nine weatherly pilot boats were wrecked or never heard from.

To those who are willing to wade through a novel with only here and there a kick in it "Sandy Hook to 62°" will appear as a gross waste of material, for in it there is a punch in every chapter.

A service that lost twenty-two vessels in twenty-two years, all through the so-called "act of God," is seagoing, rates a salute, and I give it.

GERSHOM BRADFORD.

COMMODORE DAVID PORTER, 1780-1843, by ARCHIBALD DOUGLAS TURNBULL. ILLUSTRATED. PUBLISHED BY THE CENTURY CO., NEW YORK AND LONDON. Price $3.50.

This work portrays in interesting style the Naval exploits of Commodore David Porter and also gives an intimate insight into his private life, not so well known, but nevertheless essential to form a proper estimate of the man.

The account of the part Porter played against the Barbary Pirates, a subject always so thrilling to the American youth, and his Atlantic campaign of 1812, abound in interest. Also the story of rounding the Horn and his sweep through the Pacific, magnificently executed though resulting in the loss of his beloved *Essex* after a gallant fight against a superior force, cannot fail to touch one's patriotic pride.

The book touches many matters of historic interest, not of general knowledge, and its readers will find it both entertaining and instructive.

HENDERSON B. GREGORY.

ASSOCIATION NOTES.

ANNUAL MEETING AND NOMINATION OF OFFICERS.

The Annual Meeting of the Society to nominate officers for the coming year was held at Washington, D. C., on Tuesday, October 1, 1929. The following were nominated:

For President:

Rear Admiral Harry E. Yarnell, U. S. N.

For Secretary-Treasurer:

Commander H. T. Smith, U. S. N.
Lieutenant Commander Roger W. Paine, U. S. N.

For Member of Council:

Rear Admiral C. W. Dyson, U. S. N.
Captain I. E. Bass, U. S. N.
Captain O. L. Cox, U. S. N.
Captain R. B. Adams, U. S. Coast Guard.
Commander H. S. Howard (CC), U. S. N.
Mr. J. F. Metten.
Mr. H. M. Southgate.

ANNUAL BANQUET.

The Society voted at the Annual Meeting to hold a banquet during 1930. The President has appointed the following Committee to make arrangements: Captain I. E. Bass, U. S. N., Captain H. R. Greenlee, U. S. N., Commander Harvey F. Johnson, U. S. Coast Guard, Mr. H. M. Southgate and the Secretary-Treasurer of the Society. The membership will be informed in detail as to arrangements as soon as they are completed.

MEMBERSHIP.

The following members have joined the Society since the publication of the August, 1929, JOURNAL:

NAVAL.

Barry, Ralph E., 320 Pequot St., New London, Conn.

Knowles, Herbert B., Lieutenant Commander, U. S. N.

Miles, Paul D., Ensign, U. S. N.

Olsen, S. A., Lieutenant, U. S. Coast Guard.

Reiffel, S. Jack, Ensign, U. S. N. R., 3508 Irving Park Boulevard, Chicago, Ill.

Simonson, Dale R., Lieutenant, U. S. Coast Guard.

Wilson, L. F., Lieutenant Commander, U. S. N. R., 122 South Michigan Avenue, Chicago, Ill.

CIVIL.

de Veyher, Constantin, Bureau of Construction and Repair, Navy Department, Washington, D. C.

Petroff, Peter A., Carboloy Company, Inc., 350 Madison Avenue, New York, N. Y.

Smith, Benjamin C., Standard Motor Construction Co., 180 Whiton St., Jersey City, N. J.

Squires, John, 947 The Terrace, Hagerstown, Md.

White, William C., 18 Custom House Wharf, Portland, Me.

ASSOCIATE.

Canter, G. W. G., Marine Dept., Metropolitan Vickers Electric Co., Ltd., Trafford Park, Manchester, England.

Flood, Raymond O., 2601 Filbert St., San Francisco, Calif.

Gavrilof, Ivan A., Avenida Alvear 4456 dept. F., Buenos Aires, Argentina.

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